

Three Dimensional Stress Analysis in a Steam Turbine Blade

by

Anas Hakimi bin Kori

Dissertation submitted in partial fulfillment of
the requirements for the
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Universiti Teknologi PETRONAS
Bandar Seri Iskandar
31750 Tronoh
Perak Darul Ridzuan

CERTIFICATION OF APPROVAL

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A project dissertation submitted to the

Mechanical Engineering Programme

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Approved by,

(Dr. Setyamartana Parman)

UNIVERSITI TEKNOLOGI PETRONAS

TRONOH, PERAK

November 2010

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

ANAS HAKIMI BIN KORI

ABSTRACT

The objective of this project study is to determine the stress distribution and stress concentration in a turbine blade by analyzing how stress acted on the steam turbine blades during its operation.

The stress concentration occurs when the steam turbine blade is working in extreme environment and the presence of the sharp edges in the profile of the blade localizes some stresses. If the material is unable to withstand the stress subjected on it, the blade will fracture and failure happens.

This project of study is about the stress analysis in the steam turbine blade. Manual calculations will be done with several assumptions made for the ease of calculation. Finite Element Analysis will be used to predict the area that most likely has the highest concentrated stress. All components of the turbine will be investigated to produce a set of simulation data. Using CATIA, three-dimensional model will be created and transferred to ANSYS for analysis purpose.

The outcome expected in this study is to be able to recognize the location on the turbine blade that has the highest stress concentration and further approach to reduce it.

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CHAPTER 1

INTRODUCTION

1.1 Project Background

A steam turbine is a heat engine that transforms the heat energy from superheated steam into kinetic energy in the form of turbine shaft rotation. The energy transformation process is from steam expansion through the nozzle or stator blade, and the jet from the steam expansion hit the rotor blade which in turn drives the rotor to move the turbine shaft.

This project of study is about analyzing the stress concentration of steam turbine blades. There are two important elements of energy extraction during the operation of a steam turbine. The first is the steam nozzle which is mounted on the stator and the rotor blade which is attached to the rotor. At the nozzle, the steam expands from a high pressure state towards the low pressure state, and producing the steam jet. At the rotor blade, the steam jet from the nozzle hits the rotor blade.

Blade force is produced from the difference between momentum entering and momentum exiting the blade row. The rotor blades facing the steam will carry most of the flow loading. As a result, the rotor rotates about the shaft axis at very high speed. The high rotational speed plus the high pressure, high temperature and high speed steam loadings lead to the centrifugal force and steam bending loading, thus resulting to the tensile and bending stresses on the turbine blades.

1.2 Problem Statement

Steam turbine blade is working in very extreme environment. It is working under high temperature and high pressure steam flow from the nozzles. The loading on a rotating turbine blade is composed of:

- i) Centrifugal force due to high speed rotation.
- ii) Bending force due to fluid change of momentum.

The steady-state stresses in the turbine blade are a combination of centrifugal load and the bending load due to steam force. Centrifugal stress is a function of the mass of material in the blade, blade length and speed. The component of centrifugal force acting radially outward exerts a tensile stress at the root. Sufficient cross-section area must be provided in the blade at the root and a material capable of withstanding the stress must also be provided.

Bending stress is caused by the fluid pressure difference due to the steam forces on the moving blade surface. The steam force produces thrust which acted in the axial and the tangential direction. The axial thrust is the thrust produced along the axis of the shaft while the tangential thrust is the thrust produced in the direction of blade's velocity vector.

With the blades rotating at such extreme condition, it is very important to analyse the stress acting upon the turbine blades during the design phase of the turbine. If a blade were to fail in-service, it can result in safety risks and cost thousands to repair and if the machine is not functioning up to the required design condition, it can cost hundreds to thousands of lost revenue. It is therefore very important to understand the forces and resultant stresses acting on a turbine blade. The calculation of blade stresses gives an understanding of how they behave in-service under a variety of operating conditions.

1.3 Objectives

- 1) To study, analyse and calculate the stresses that acted on the single-stage impulse steam turbine blades.
- 2) To do the stress analysis by Finite Element Method by using computer software and compare the result with the calculated values.

1.4 Scope of Study

The scope of this study is to do stress analysis on single-stage impulse steam turbine. Research about stress acted on the steam turbine blade will be carried out. It is important to be able to calculate the stress using hand calculation before proceeding to use the computer software.

For the computer tools, first the turbine blade is modeled using CATIA V5. From there, the model can be exported to ANSYS for the stress analysis purpose. The operating condition will be simulated using ANSYS. Once the result is obtained, it will be compared to the hand calculation results.

CHAPTER 2

LITERATURE REVIEW

2.1 Stress Analysis of Steam Turbine Blades

Stress analysis is an engineering discipline that determines the stress in materials and structures subjected to static and dynamic forces or loadings.

The objective of stress analysis process is to determine whether the element or structure, can safely withstand the specified loadings when the structure is subjected to it. This is achieved when the resultant stresses due to the applied loadings to the material are less than the allowable stresses the material can withstand. Allowable stress is the working stress that happened to a structure. The allowable stress must be lower than the failure stress of the material. Failure stress is found from experimental testing of the material.

The relationship between these working stress and failure stress is determined by Safety Factor. Safety factor is a design requirement for a structure based on the uncertainty in loads, material strength, and consequences of failure (note that safety factor is the minimum required factor for design, while Design Factor could be equal or exceed the safety factor). Safety factor is selected based on experience so that the above mentioned uncertainties are accounted for when the member is used under similar conditions of loading and geometry. Generally, safety factor is the ratio of material strength to the design loads. In this case, it is the ratio of failure stress to the allowable stress. However, Safety Factor will not be covered as it is not one of the interests in this study.

Stress concentration is a location where stress is concentrated. An object is strongest when force is evenly distributed over the surface area. The reduction in acting surface area results in the localized increase of stress. This is logical with the definition of stress which is the intensity of loads or forces when subjected to a unit of area. Simply put, it is:

$$\text{Stress, } \sigma = \frac{\text{Force}}{\text{Area}} \quad (\text{E 2.1})$$

The main cause of the stress concentration is due to the geometric discontinuities that increases the intensity of the buildup of the stress field. In engineering practice, the actual stress distribution does not have to be determined. Instead, the maximum stress at these sections must be known. For design or analysis, it is necessary to determine the maximum stress acting on the smallest cross-sectional area.

To determine the stress concentration on the steam turbine blade, it is very important to analyse the loadings that acted on the blade. The loadings that acted on the turbine blade are **centrifugal forces, fluid-pressure differences** and **vibration**. The severest stress is imposed by the centrifugal forces due to high rotating speed. Fluid-pressure differences is due to the gust forces which are exerted by the working fluid (the hot, fast and high pressure steam flow) on the moving blade surface and resulted to the bending stress. Vibration is neglected for the ease of this study. Therefore, the determined stresses exerted to steam turbine blades are **centrifugal stress** and **bending stress**.

The steady-state stress at any section of a parallel sided blade is a combination of direct tensile load due to the centrifugal force and bending load due to the steam force. The combined tension and bending stresses is maximum at blade roots and diminishes with radius.

2.2 KKK AFA 4 Steam Turbine Blades

In this study, Kuhnle, Kopp & Kausch (KKK) type AFA 4 Steam Turbine will be analysed. The KKK AFA 4 Steam Turbine is the turbine that is used in a Kg Gajah Palm Oil Mill, Kg Gajah, Perak. This type of turbine is the single-stage impulse turbine.



Figure 2.1: Turbine, facing north



Figure 2.2: Turbine, facing south

The operating data of the KKK AFA 4 steam turbine is as follows:

Coupling Output	Maximum	1265	kW
Output speed	Normal	1500	1/min
Speed Trip	Minimum	1678	1/min
	Normal	1695	1/min
	Maximum	1712	1/min
Gear ratio			8.21
Live steam pressure	Minimum (limit)	15.50	bar g
	Normal (operating)	19.60	bar g
	Maximum (limit)	23.70	bar g
Live steam temperature	Minimum (limit)	203.00	°C
	Normal (operating)	214.00	°C
	Maximum (limit)	242.00	°C
Exhaust steam pressure	Minimum (limit)	2.50	bar g
	Normal (operating)	3.50	bar g
	Maximum (limit)	4.50	bar g
Exhaust steam temperature	Maximum (limit)	180	°C

Table 2.1: Operating data and limits of KKK AFA 4 steam turbine

2.3 Material of the Turbine Blade

Based on the service instructions of the KKK Steam Turbine type AFA 4, the material for the turbine blade is **X22CrMoV12.1**, which is the equivalent for Martensitic Steel No. 1.4923. The blade material is the same with the rotor material

as the blades themselves are fixed homogenously with the rotor wheel. The chemical composition of material is shown in Table 2.2. Table 2.3 below shows the material properties of the X22CrMoV12.1.

Summary of properties: For use up to 580°C. Also used for turbine parts.

C	Si	Mn	P	S	Cr	Mo	Ni	V
0.20	0.30	0.60	≤0.035	≤0.035	11.80	1.00	0.60	0.30

Table 2.2: Chemical composition of X22CrMoV12.1

PROPERTIES	VALUE
Density (kg/m ³)	7700
Yield Strength (MPa)	≥600
Tensile Strength (MPa)	800-950
Modulus of Elasticity (MPa)	200 000
Poisson's Ratio	0.29
Elongation (%)	14
Reduction of Area (%)	40
Electrical Resistivity, 20°C (Ω mm ² /m)	0.60
Magnetizable	Yes
Thermal conductivity, 20°C (W/m K)	24
Specific Heat Capacity, 20°C (J/ kg K)	460
Thermal expansion coefficient (µm/mK)	Between 20 – 200°C = 11.1

Table 2.3: Material properties of X22CrMoV12.1

2.4 Principles of the Steam Turbine Blading

Two principles are used to design and construct steam turbines to get the desired result. These are the reaction turbine and the impulse turbine. Both methods convert the thermal energy stored in the steam into useful work, but they differ somewhat in the way they do it. In the following paragraphs is the explanation of the two basic turbine principles.

The reaction blading principle depends on the blade diverting in the steam flow and gaining kinetic energy by the reaction. For the reaction blading turbine, the steam pressure drop is divided between the fixed and moving blades.

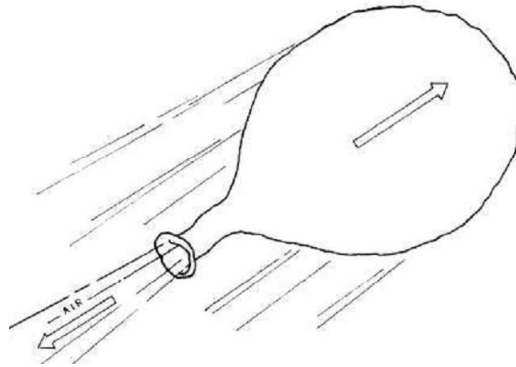


Figure 2.3: Demonstration of the reaction turbine

The impulse blading principle is that the steam is directed at the blades and the impact of the steam jets drives the blades around. For the impulse blading type, the whole of the stage pressure drop takes place in the fixed blade (nozzle) and the steam jets act on the moving blade by impinging on the blades.

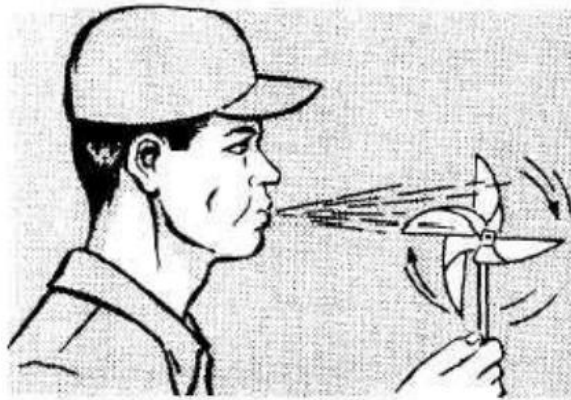


Figure 2.4: Demonstration of the impulse turbine

The impulse blade is subjected to bending from the tangential force exerted by the fluid. Meanwhile, the reaction blade has an additional force resulting from the large axial thrust because of the pressure drop which occurs in the blade passages.

2.5 Centrifugal Stress

One of the loads that acted on the turbine blade is centrifugal force. Centrifugal stresses are a function of the mass of material in the blade, blade length and speed.

The component of centrifugal force acting radially outward exerts a tensile stress at the root. This direct tensile stress (centrifugal stress) is maximum at the blade root. Following are the calculation method to get the centrifugal force on the blade. The general equation for centrifugal force is

$$F = m\omega^2 r \quad (\text{E 2.2})$$

where m is the mass of the object, r is the radius of the curvature and ω is the angular or rotational speed of the object.

Note that the mass of the blade is distributed over its length and the radius of curvature also changes along the length of the blade. Consider a small segment mass δm having width δr at a distance r from the center. Then the equation for the centrifugal force is δF is given by

$$\delta F = \delta m \omega^2 r \quad (\text{E 2.3})$$

Practically, a turbine blade tapers geometrically in thickness from the root towards its tip. However for simplicity, it is assumed that the blade is having a constant cross sectional area and material density. These assumptions lead to the integral of δm will be directly proportional to the δr , that is

$$\delta m = \rho A \delta r \quad (\text{E 2.4})$$

Then equation E x.x will become

$$\begin{aligned} \delta F &= \delta m \omega^2 r \\ \delta F &= \rho A \omega^2 r \delta r \end{aligned} \quad (\text{E 2.5})$$

Equation E x.x can be formally as

$$dF = \rho A \omega^2 r dr \quad (\text{E 2.6})$$

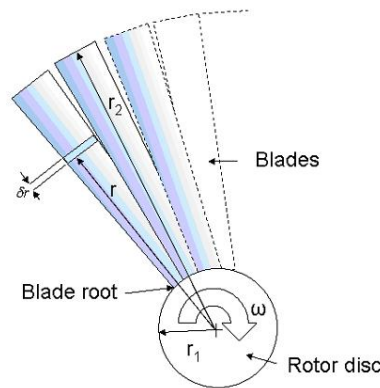


Figure 2.5: Simplification of blade dimensions

The area of interest now is to have the centrifugal force acting on the turbine blade from the blade root towards its tip. Integration of equation (E 2.6), with limits from r_1 to r_2 would obtain

$$F = \rho A \omega^2 \int_{r_1}^{r_2} r dr$$

$$F = \rho A \omega^2 \left(\frac{r_2^2 - r_1^2}{2} \right) \quad (\text{E 2.7})$$

Once centrifugal force is obtained, I will proceed to the next step which is estimating the nominal stress σ on the blade root using the relation

$$\sigma = \frac{F}{A_{\text{root}}} \quad (\text{E 2.8})$$

2.6 Fluid Pressure Difference

Fluid pressure differences is due to the gust forces which are exerted by the working fluid (the hot, fast and high pressure steam flow) on the moving blade surface and resulted to the bending stress. The gust force, or the force from the steam flow produces thrust which acted in the axial and the tangential direction. The axial thrust is the thrust produced along the axis of the shaft while the tangential thrust is the thrust produced in the direction of blade's velocity vector.

As mentioned in section 2.2, KKK AFA4 Steam Turbine is the impulse type steam turbine. In impulse turbine, the whole stage pressure drop takes place in the fixed blade (nozzle). The steam jet impinges on the blade, and the impact drives the blades around. The tangential thrust load exerted by the steam produces the fluid pressure difference. Simply put, the graphical presentation of the fluid pressure difference is as follows:

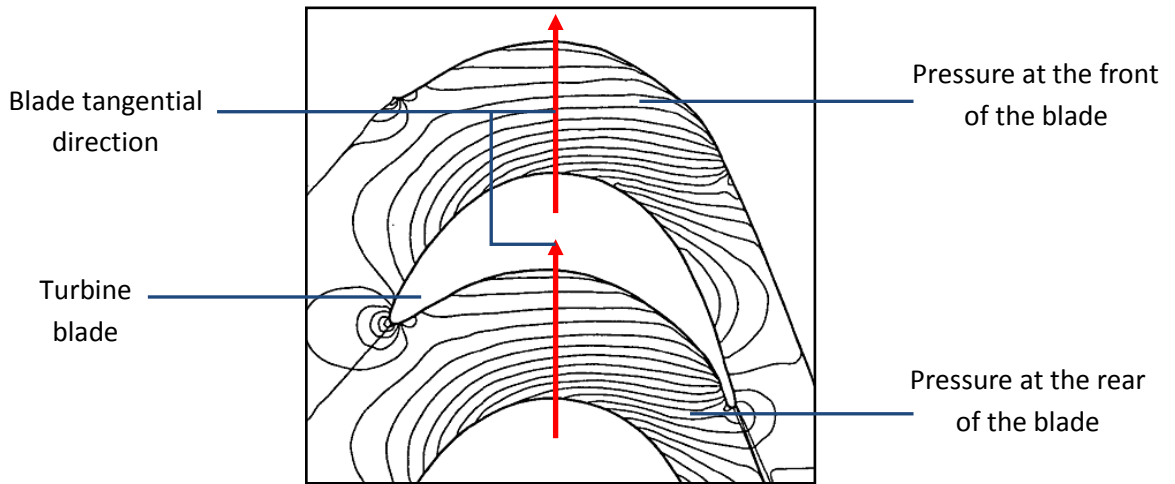


Figure 2.6: The graphical representation of the fluid pressure difference

The impact from steam jet resulted to the greater pressure at the rear of the blade than the pressure at the front of the blade. The steam jet exerts tangential force to the impulse blade and produces bending to the blade. The moment of this force stress due to this are both maximum at the blade root.

The tangential force impressed by the steam jets on the blades are given by equation

$$F_t = \dot{m} \Delta V_\omega \quad (\text{E 2.9})$$

where

F_t = tangential force of the steam jet

\dot{m} = mass flow rate of the steam

ΔV_ω = change in velocity in tangential direction

To measure the fluid pressure difference of a steam turbine blade requires the understandings of the velocity diagrams at the blade itself. First is the blade orientation, steam direction when it leaves the nozzle until it hits the blade nose and when the steam leaves the throat.

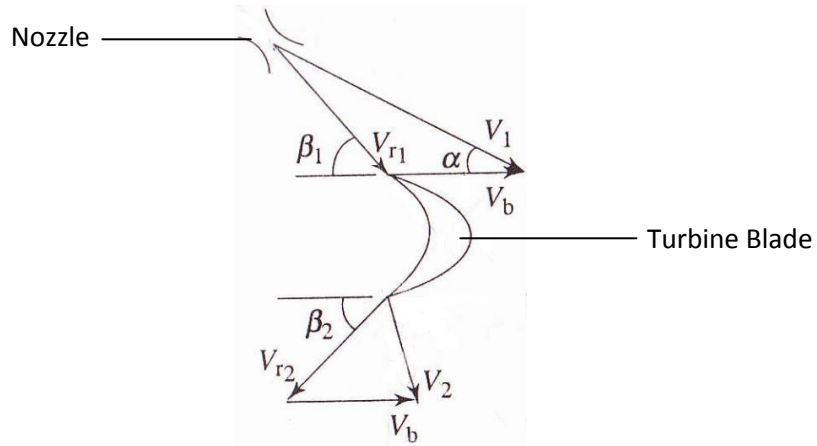


Figure 2.7: The velocity diagram of steam entering and exiting the turbine blade

In the diagram,

V_1 and V_2 are the steam absolute inlet and exit velocities respectively;

V_{r1} and V_{r2} are the inlet and exit steam relative velocities (to the blade);

V_b is the tangential blade velocity;

α is the absolute steam inlet angle;

β_1 and β_2 are the inlet and exit angles of relative velocities.

Taking the tangential blade velocity, V_b as reference, the two velocity diagrams can be superposed as in this figure 2.8 below:

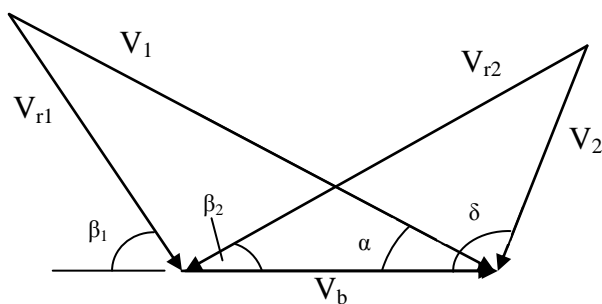


Figure 2.8: The velocity diagram of steam entering and exiting the turbine blade

In the figure above, the angle between the tangential blade velocity and the absolute steam exit velocity, δ is introduced.

Resolving both V_1 and V_2 to the tangential and axial axis, the equation will be

$$\Delta V_t = V_1 \cos \alpha - V_2 \cos \delta \quad (\mathbf{E\ 2.10})$$

$$\Delta V_a = V_1 \sin \alpha - V_2 \sin \delta \quad (\mathbf{E\ 2.11})$$

Now, the loading will be

$$F_t = \dot{m} (V_1 \cos \alpha - V_2 \cos \delta) \quad (\mathbf{E\ 2.12})$$

$$F_a = \dot{m} (V_1 \sin \alpha - V_2 \sin \delta) \quad (\mathbf{E\ 2.13})$$

Having identified all the three loadings the turbine blade is subjected to, the orientation of the loading is also very important to be understood. The image below shows the orientation of the loadings. The axial force is directed across the blade, the tangential force is acted to push the blade forward while the rotational load tend to rotate the blade about the shaft axis.

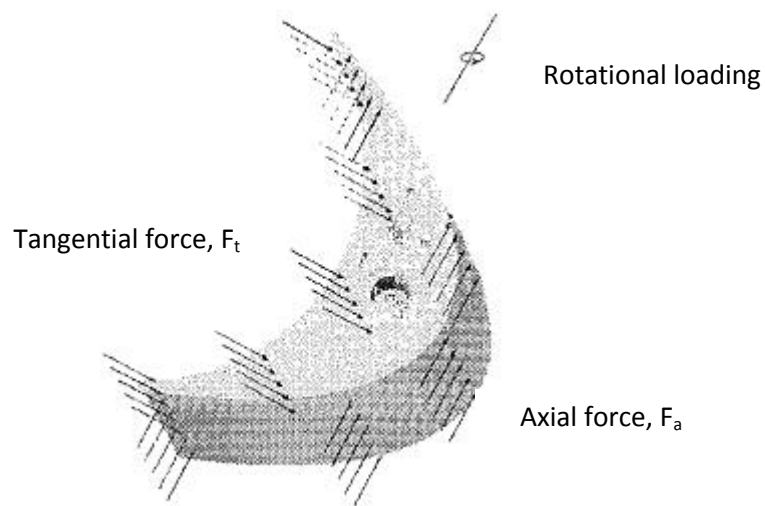


Figure 2.9: The orientation of the steam forces and the rotation of rotor

2.7 Finite Element Analysis and ANSYS

The finite element method is a numerical procedure that can be used to obtain solutions to a large class of engineering problems involving stress analysis, heat transfer, electromagnetism, and fluid flow.

There are three general procedures in FEA which are Preprocessing, Solution and Post-processing.

The Preprocessing phase involves defining the geometric domain of the problem, the element types used and their material properties, the geometric properties of the elements, the element connectivity, the physical constraints and also the loadings. In this phase, equations for the elements will be developed. This Preprocessing phase is the most important phase as it determines the entire result of the analysis.

In Solution phase, a set of the linear or nonlinear equations are solved simultaneously to obtain the nodal result.

In Post-processing phase, the results will be analyzed and evaluated. The examples of the evaluation are equilibrium check, animate dynamic model behavior, and produce color-coded temperature plots.

ANSYS is a general-purpose finite element modeling package for numerically solving a wide variety of mechanical problems including the stress acted on steam turbine blades. Generally, the process of simulation can be divided into these three following steps which is just the same as the general procedures of FEA mentioned above:

1) Preprocessing: Defining the problem

The turbine blade area, volumes type and material properties are defined.

2) Solution: Assigning loads, constraints and solving

The loads which are the forces calculated are applied on the turbine blade after setting its constraint which is its root.

3) Post-processing: Further processing and viewing the results

The stress contour diagrams are produced and viewed for further analysis.

CHAPTER 3

METHODOLOGY/PROJECT WORK

3.1 Project Flow Chart

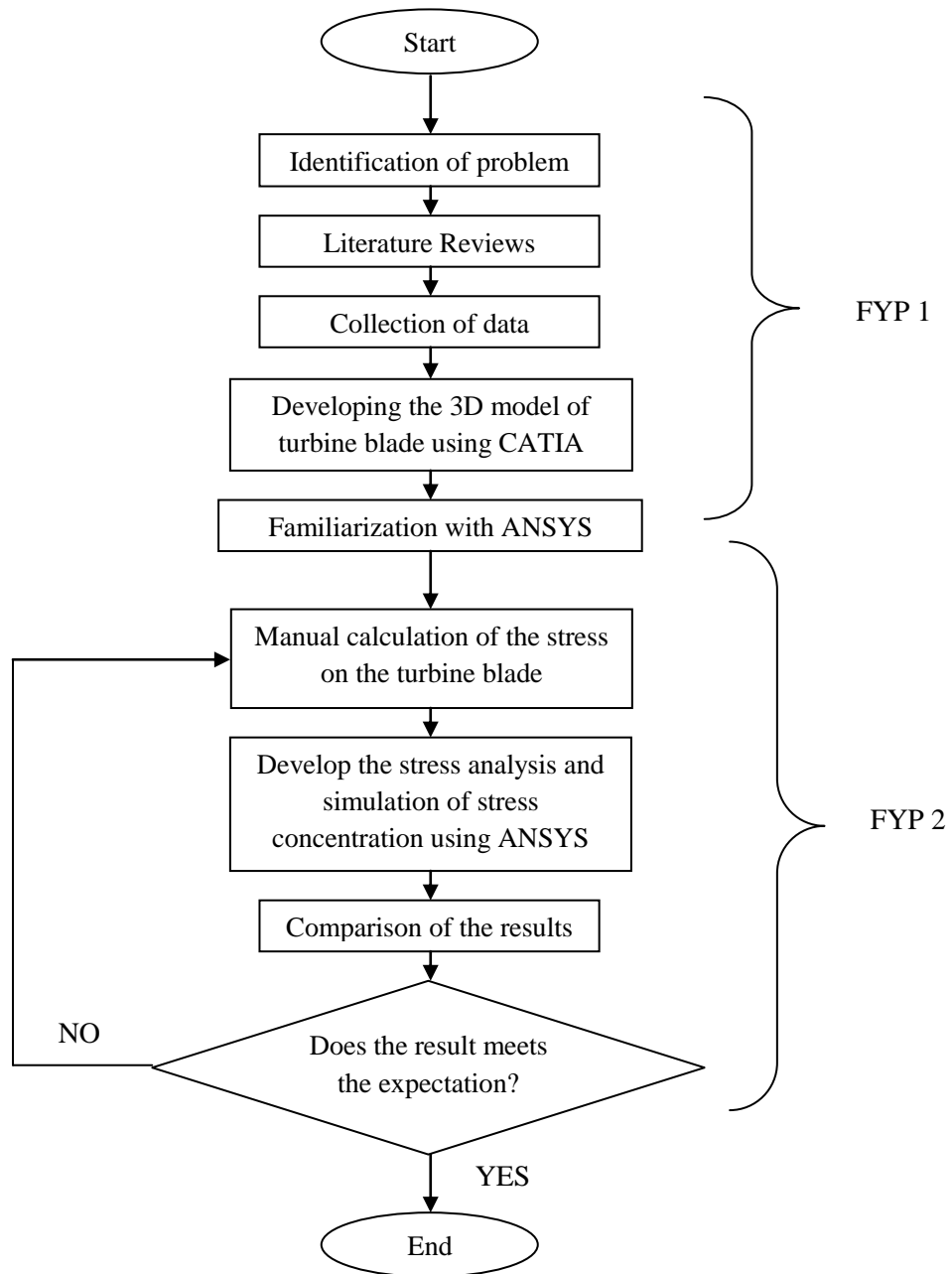


Figure 3.1: Project Flow Chart

3.2 Tools/Equipment Required

The tools and equipment that are required in this project are a Microsoft-Windows-based computer together with the Microsoft Office, CATIA and ANSYS softwares. Microsoft Office programs are Microsoft Word for writing reports and Microsoft Excel for rearranging data. CATIA is for three dimensional model drafting of the turbine blade and also to aid with the geometrical problems. ANSYS is a Finite Element Analysis software that will be used for the model simulation. The model is imported from CATIA which has been mentioned earlier.

3.3 Developing the 3D Solid Model of the Turbine Blade

In developing the 3D model of the turbine blade, CATIA will be used. CATIA stands for Computer Aided Three-Dimensional Interactive Application is a multiplatform CAD/CAM/CAE commercial software suite. CATIA is widely used throughout the engineering industry, especially in the automotive and aerospace sectors. CATIA is chosen because of the 3D modeling capability and its integrability to be exported to ANSYS for the purpose of stress analysis and simulation. For the turbine blade model, the KKK AFA 4 single-stage impulse steam turbine is used. The measurement of the blade dimension is done during the Kg. Gajah Palm Oil Mill shut down of the steam turbine for maintenance purpose.

3.4 Collection of Required Data

Collection of data is required before manual calculation and computer simulation are taking place. The required data includes the operating parameter of the KKK AFA 4 steam turbine from the turbine manual as well as the calculation methods and formulae of the stress that acted on the steam turbine blade from books, research paper and internet.

3.5 Familiarization with ANSYS

ANSYS will be used for the computational way of measuring stress that acted on the turbine blade. ANSYS also will be used as the simulation tool to inspect the stress distribution and stress concentration on the turbine blade. For that purpose, ANSYS has to be familiarized first. ANSYS familiarization is done by using the model developed earlier from the CATIA model.

3.6 Manual Calculation of the Turbine Blade Stress

The manual calculation includes the calculation of centrifugal stress and the fluid pressure difference. The centrifugal stress is caused by the high speed rotation of the rotor and results in the tensile loadings to the turbine blades. Fluid pressure difference results in the bending stress at the root of the blade. The stress acted on the steam turbine blade is maximum at the root section and diminished with the blade height.

There are many steps for manual calculation of the stresses acted on the turbine blades. To ease the calculation, one needs to consider developing the free body diagram of the turbine blades. One single blade will be taken as a model with loads is acting on it. The loads are centrifugal force and steam force. The steam force can be resolved into axial force, which acts along the shaft axis and tangential force, which acts on the whirl direction of the rotor.

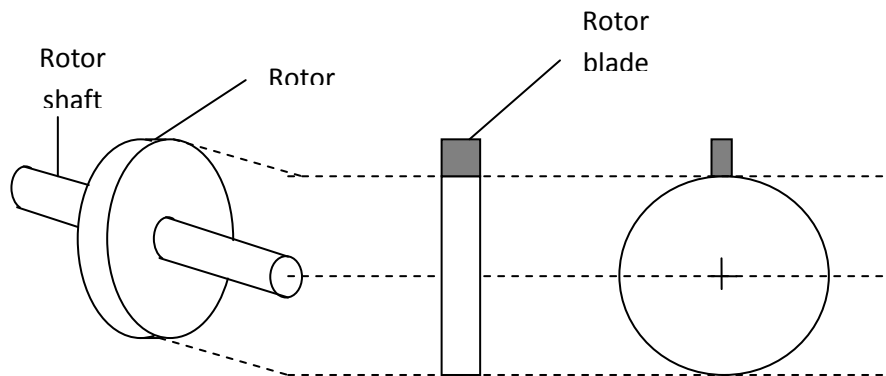


Figure 3.2: Rotor mounted on shaft

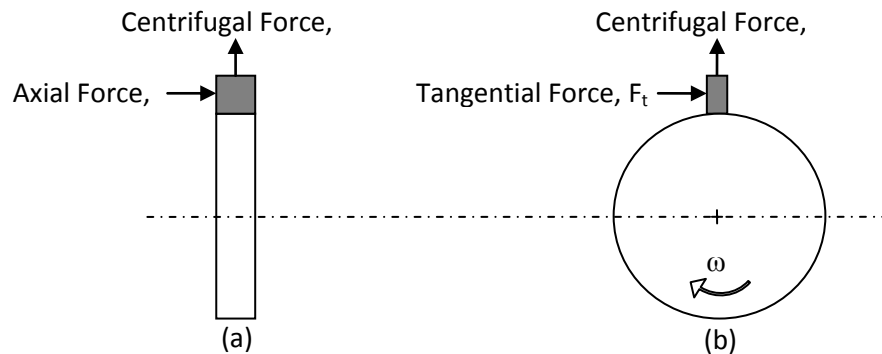


Figure 3.3: Figure (a) is the front view of the rotor, which shows the orientation of the axial force while figure (b) is the side view of the rotor which shows the orientation of the tangential force

3.6.1 Centrifugal Stress

Centrifugal force is the force that is resulted from the circular motion of the rotor. The force is directed outward from the center of the rotation, and resulting to the tensile loading on the turbine blades. The centrifugal stress, which is caused by the centrifugal force is more important than other stress exerted by flow forces since its value is dominant.

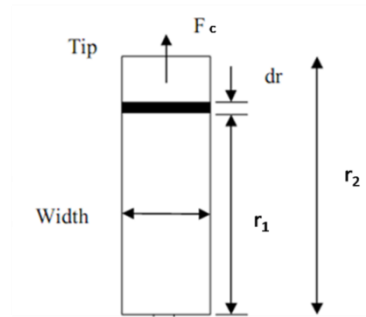


Figure 3.4: Free body diagram of rotor blade showing the centrifugal force

Having the required information, centrifugal force can be calculated using equation 2.7, then the centrifugal stress calculation can be proceeded using equation 2.8.

$$F_c = \rho A_{cross-section} \omega^2 \left(\frac{r_2^2 - r_1^2}{2} \right) \quad (\text{E 2.7})$$

$$\sigma_c = \frac{F_c}{A_{root}} \quad (\text{E 2.8})$$

3.6.2 Fluid Pressure Difference (Steam Bending Stress)

Steam bending stress is caused by the axial and tangential loadings acted on the turbine blades. The projection of the forces is depicted as below:

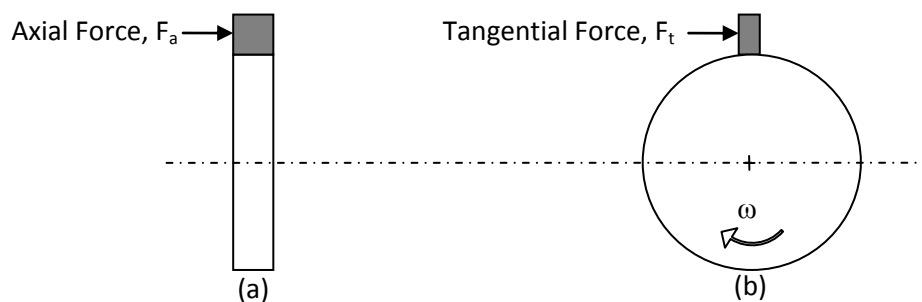


Figure 3.5: Front and side views of the rotor showing the direction of axial and tangential forces

There are several steps for doing the calculation. The steps are:

- 1) Draw the free body diagram of the turbine blade.
- 2) Draw the velocity diagram of the steam.
- 3) Once, the velocity diagram is constructed, axial and tangential force is calculated.
- 4) Calculate the moment caused by the each force calculated.
- 5) Convert the moment direction in the axis that is across the horns (blade's tips) and perpendicular to a line across the horns.
- 6) Bending stress can now be calculated.

To calculate the steam loadings, the velocity of steam is required. This input data can be obtained by employing the velocity diagram of steam at turbine blade. Using the inlet and exit velocities from the velocity diagram, axial and tangential force can be obtained by employing equations 2.12 and 2.13.

$$F_t = \dot{m} (V_1 \cos \alpha - V_2 \cos \delta) \quad (\mathbf{E\ 2.12})$$

$$F_a = \dot{m} (V_1 \sin \alpha - V_2 \sin \delta) \quad (\mathbf{E\ 2.13})$$

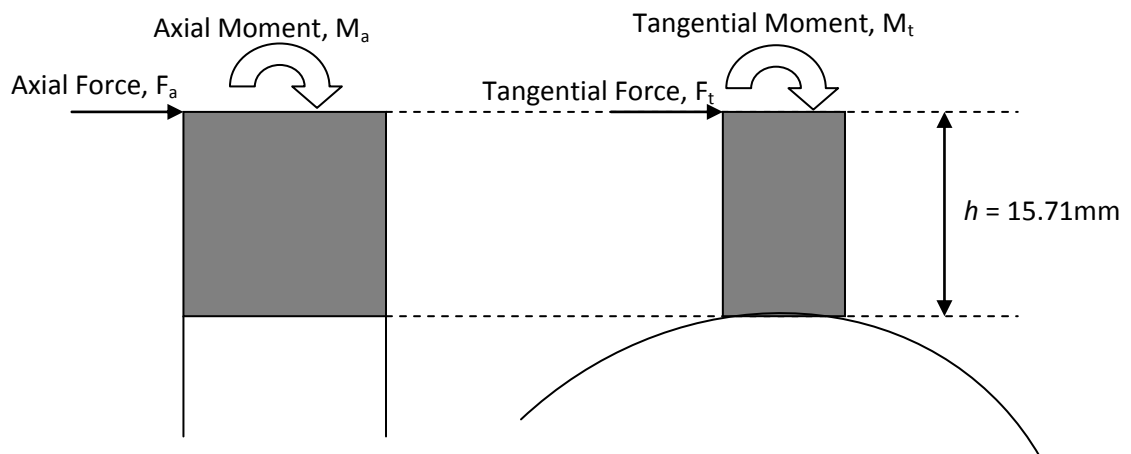


Figure 3.6: The axial and tangential moments caused by the corresponding loadings

When the forces are known, having the moment arm, which is the blade length, the corresponding moments can be calculated.

$$M_t = F_t \cdot h \quad (\text{E 3.1})$$

$$M_a = F_a \cdot h \quad (\text{E 3.2})$$

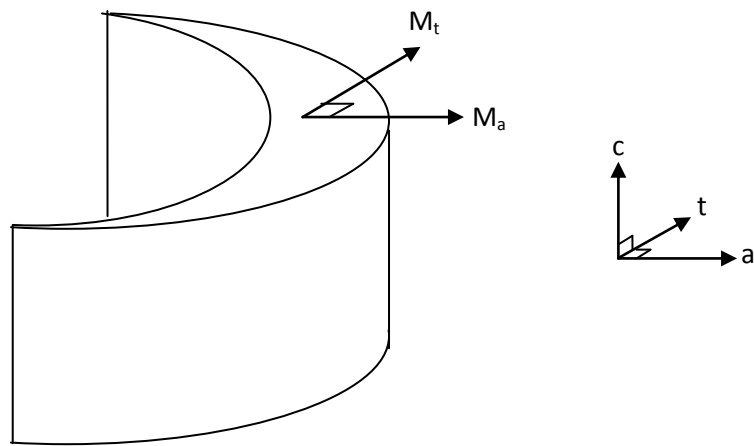


Figure 3.7: The direction of tangential and axial moments

When the tangential and axial components of the bending moments have been determined, the actual bending moments in the cane, which is across the horns and perpendicular to a line across the horns, which is the direction in which the section moduli are normally calculated can be established for a vane with a setting angle of θ as shown in Figure 3.8 . These moments can be resolved in the directions XX and YY as M_{xx} and M_{yy} to give equations 3.3 and 3.4.

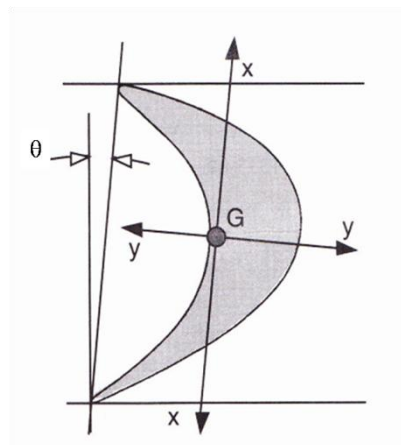


Figure 3.8: Vane section for steam bending moments

$$M_{xx} = M_t \cos\theta + M_a \cdot \sin\theta \quad (\text{E 3.3})$$

$$M_{yy} = M_t \cdot \sin\theta + M_a \cdot \cos\theta \quad (\text{E 3.4})$$

When the actual bending moment is obtained, then the bending stress calculation can now be done. The stress calculation requires the value of the moment of inertia and also the length from centroid to the point of interest, with respect to the *XX* and *YY* directions.

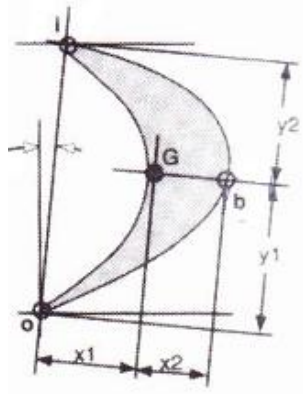


Figure 3.9: Figure shows the length from centroid to the point of interest with respect to *XX* and *YY* directions, noted as x_1 , x_2 , y_1 and y_2

The bending stress is calculated using equation:

$$\sigma = \frac{M_{xx}}{I_{yy}} x \quad (\text{E 3.5})$$

$$\sigma = \frac{M_{yy}}{I_{xx}} y \quad (\text{E 3.6})$$

I_{xx} and I_{yy} is the area moment of inertia of the surface while x and y are the distance from centroid, depending on the point of interest. In this project, the inlet i , exit o and the front blade b , sides are chosen.

3.7 Simulation of the Stress Concentration by ANSYS and Comparison of Results between Manual Calculation and Computer Simulation

The stress analysis of the turbine blades will be conducted by using ANSYS. ANSYS is a software that can develop a general-purpose finite element analysis and computational fluid dynamics. By using ANSYS, results on how the stress affects the turbine blade can be obtained. The types of stress that are acting towards the blades also can be identified by ANSYS. The result will show the most vulnerable part of the blade when there is high centrifugal load and high pressure difference acting towards the blade. With the result, a better design of the turbine blade could be

achieved in the future. To achieve the result, the real operating condition will be simulated in ANSYS.

3.7.1 Meshing in ANSYS

Meshing is the process of ‘sub-divide’ the three dimensional model into elements, connected at nodes. The idea is to build a complicated object with simple blocks, or, dividing a complicated object into small and manageable pieces. The finer the element is, that is the smaller the element is, the more accurate the result will be.

Firstly, the three dimensional model is imported from CATIA so that the solid model can be used in ANSYS. Then, material properties of X22CrMoV12.1 are specified using the ‘Engineering Data’ option menu, as per Figure 3.10. Next the element is chosen. In this case, the element Hexahedral Dominant method (Hex Dominant) with size of 1mm, as per Figure 3.11 and Figure 3.12 is chosen for meshing purpose. Depending on the model complexity, desired mesh quality and type, and how much time a user is able to spend meshing, a user has a scalable solution to generate a quick automatic hex or hex dominant mesh, or a highly controlled hex mesh for optimal solution efficiency and accuracy.

Structural Steel	
Structural Add/Remove Properties	
<input type="checkbox"/> Young's Modulus	2.e+011 Pa
<input type="checkbox"/> Poisson's Ratio	0.29
<input type="checkbox"/> Density	7700. kg/m ³
<input type="checkbox"/> Thermal Expansion	1.11e-005 1/°C
<input type="checkbox"/> Alternating Stress	
<input type="checkbox"/> Strain-Life Parameters	
<input type="checkbox"/> Tensile Yield Strength	6.e+008 Pa
<input type="checkbox"/> Compressive Yield Strength	6.e+008 Pa
<input type="checkbox"/> Tensile Ultimate Strength	9.e+008 Pa
<input type="checkbox"/> Compressive Ultimate Strength	0. Pa
Thermal Add/Remove Properties	
<input type="checkbox"/> Thermal Conductivity	24. W/m·°C
<input type="checkbox"/> Specific Heat	460. J/kg·°C
Electromagnetics Add/Remove Properties	

Figure 3.10: Material properties

Details of "Hex Dominant Method" - Method	
Scope	
Scoping Method	Geometry Selection
Geometry	1 Body
Definition	
Suppressed	No
Method	Hex Dominant
Element Midside Nodes	Use Global Setting
Control Messages	No

Figure 3.11: Element types

Details of "Body Sizing" - Sizing	
Scope	
Scoping Method	Geometry Selection
Geometry	1 Body
Definition	
Suppressed	No
Type	Element Size
Element Size	1.e-003 m
Edge Behavior	Curv/Proximity Refinement

Figure 3.12: Element size, 1mm

3.7.2 Simulation of Stress by ANSYS

For this project, three simulations will be carried out. The simulations are the centrifugal stress which is based on the rotation of the rotor, the steam bending stress which is based on the steam forces, and the combination of centrifugal stress and steam bending stress.

i) Centrifugal Stress

In ANSYS, the model will be fixed at the root face, and rotational velocity will be applied. The location of the rotational velocity applied will be specifically the center of the rotor. The rotational velocity will be calculated based on the operating specification of the turbine.

ii) Steam Bending Stress

The simulation will be done by specifying the loads applied to the blade. The loads is specific with its direction acting toward the tangential and the axial direction.

iii) Combination of the centrifugal stress and steam bending stress

Once all the loads are identified, all the inputs including the centrifugal load and steam bending load will be applied to the model in ANSYS.

When the results are obtained, the result will be compared to the solutions obtained from the manual calculation method.

3.8 Gantt Chart

Please refer to the Appendices for the Gantt Charts of FYP 1 and FYP 2.

CHAPTER 4

RESULTS AND DISCUSSIONS

4.1 Result

4.1.1 Three Dimensional Model of the Turbine Blade

This model is based on the turbine blade of KKK AFA 4 Steam Turbine. The turbine is located in Kg Gajah Palm Oil Mill and is opened for maintenance on 1st May 2010. The rotor and turbine blade are measured to get its dimension for modeling in CATIA.



Figure 4.1: General view of the turbine rotor

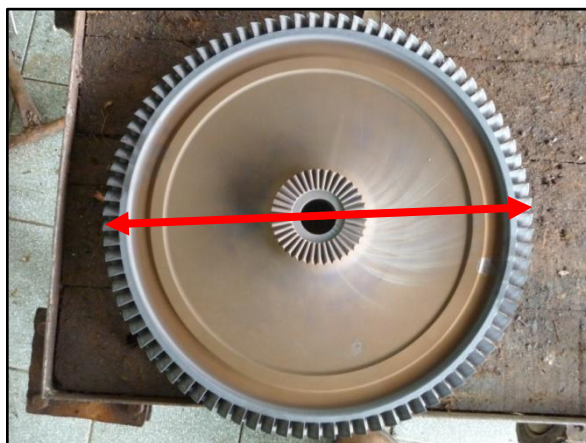


Figure 4.2: The dimension of the rotor as measured

Then, the dimension of the rotor blade is the taken using Vernier Calliper.



Figure 4.3: The top view of the rotor blade

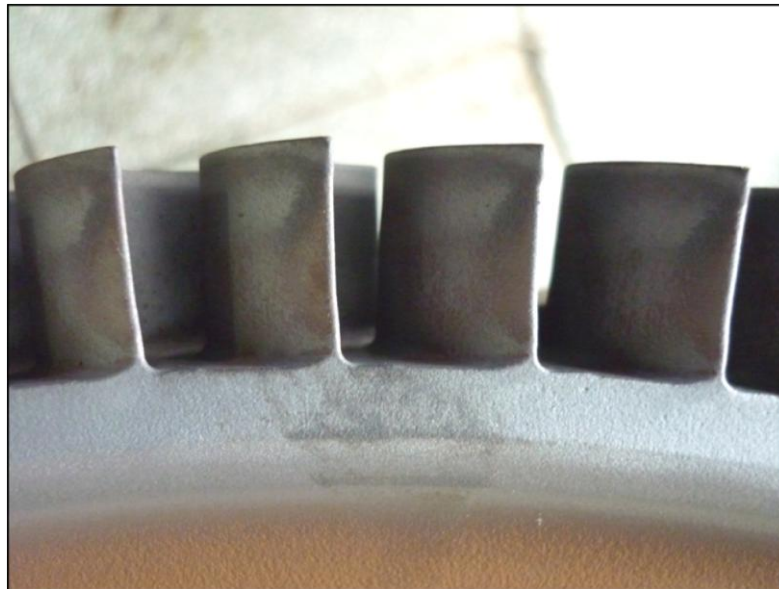


Figure 4.4: The side view of the rotor blade

The dimension of the rotor blade is the taken using Vernier Calliper. The blade profile dimension is given as below:

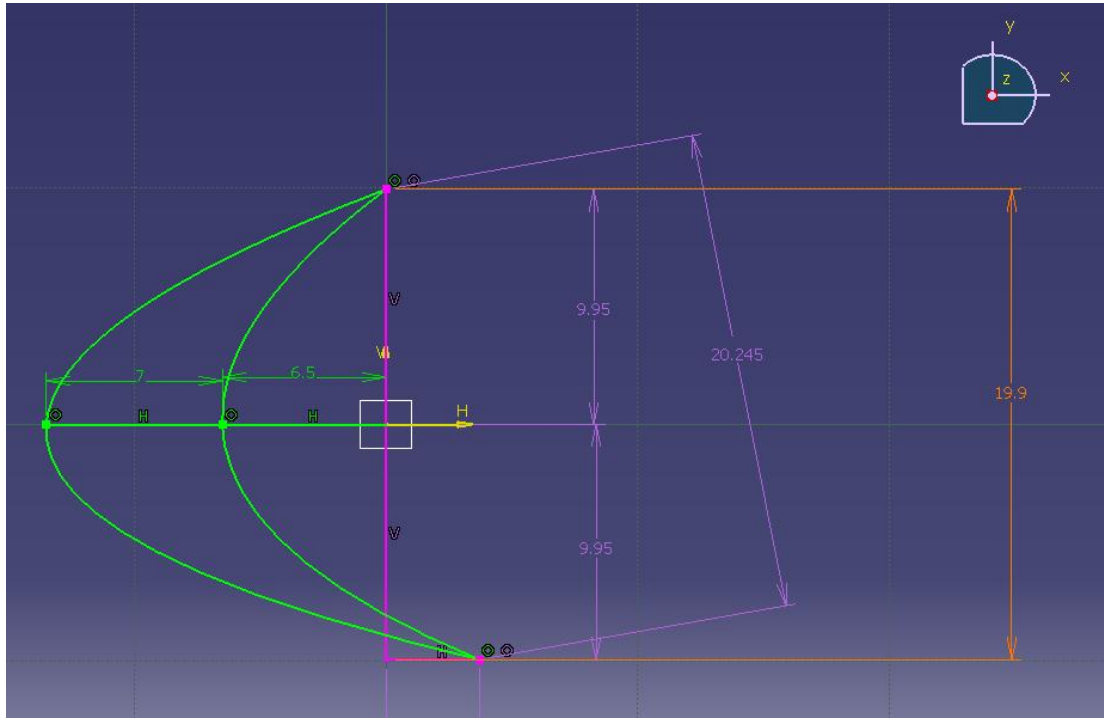


Figure 4.5: The blade profile and its dimension in millimeter

Below are several dimensions of the blades:

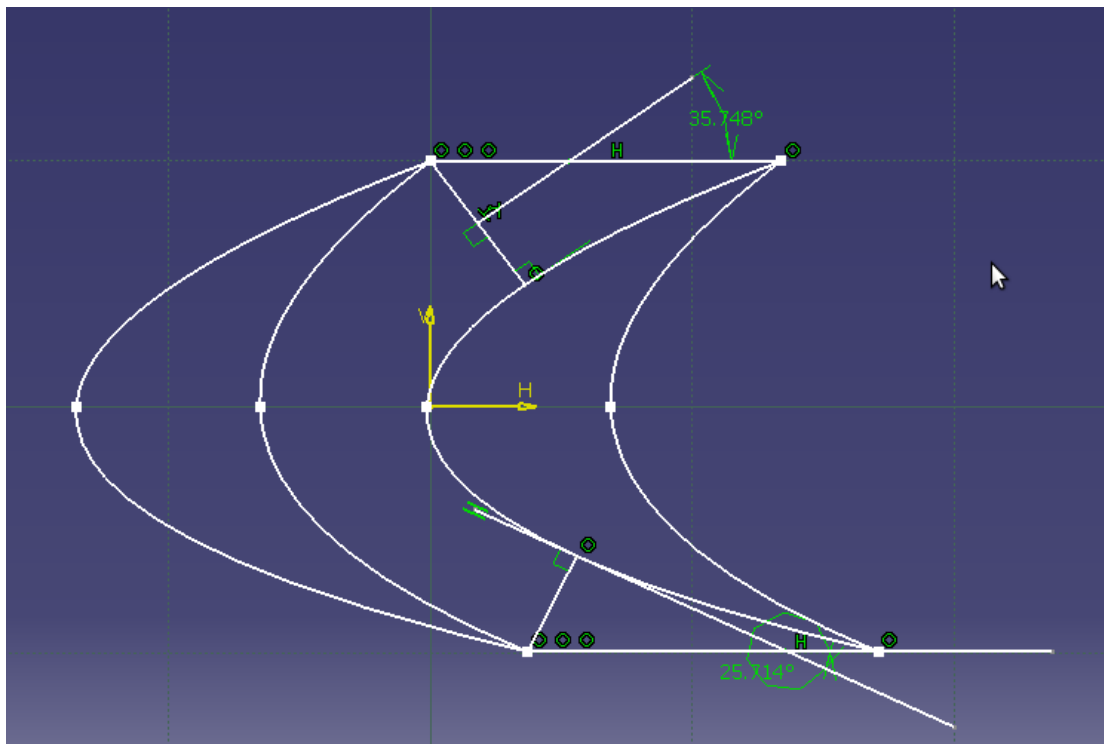


Figure 4.6: Using CATIA to measure the inlet and exit steam relative velocity vector, β_1 and β_2

The average height of the blade is 15.71mm. The measurement is taken three times and the average is calculated. The height can be referred to Figure 4.4.

Height 1	15.51 mm
Height 2	15.80 mm
Height 3	15.82 mm
Average height	15.71 mm

Table 4.1: Height measurement for the blade

Though the nose and the tail of the blade are not aligned, but the thickness still can be measured by taking the axial length of the rotor. The actual length is 20mm and the measured is 19.9mm. We will be using the axial length as 19.9mm throughout this study.

The CATIA model used in this project is as follows:

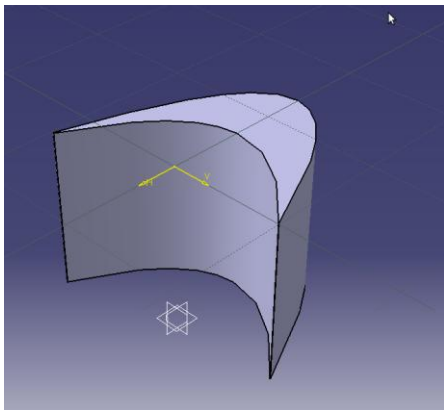


Figure 4.7: Isometric view

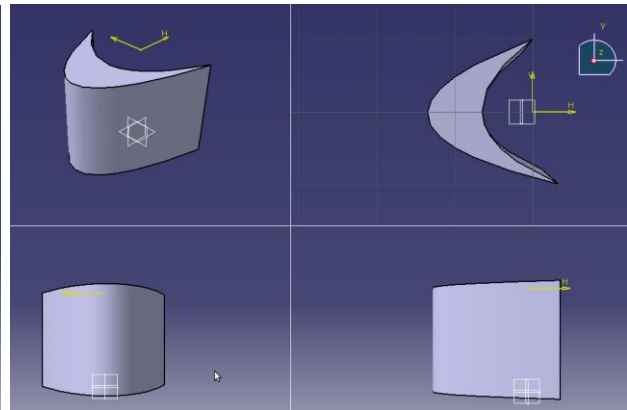


Figure 4.8: Multi-view

4.1.2 Manual Stress Calculation Acted on the Steam Turbine Blade

4.1.2.1 Calculation of the Centrifugal Stress

Outer radius of rotor, r_2	0.2066m
Inner radius of the rotor, r_1	0.19089m
Turbine output speed	1500 rpm
Gear ratio	8.21
Cross-section Area, $A_{\text{cross-section}}$	$9.343 \times 10^{-5} \text{ m}^2$
Root Area, A_r	$9.343 \times 10^{-5} \text{ m}^2$
Blade density, ρ	7700 kg/m^3

Table 4.2: The data required for the centrifugal force calculation

The rotational speed, is measured by having the output operating speed and the gear ratio.

$$\begin{aligned}
 \text{Rotational speed of pinion shaft} &= \text{gear ratio} \times \text{output speed} \\
 &= 1500 \times 8.21 \\
 &= 12315 \text{ rpm} \times 2\pi/60 \\
 &= \underline{1289.62 \text{ rad/s}}
 \end{aligned}$$

$$\begin{aligned}
 F_c &= \rho A_{\text{cross-section}} \omega^2 \left(\frac{r_2^2 - r_1^2}{2} \right) \\
 &= (7700 \text{ kg/m}^3) (9.343 \times 10^{-5} \text{ m}^2) (1289.62 \text{ rad/s})^2 \left(\frac{(0.2066)^2 - (0.19089)^2}{2} \right) \text{ m}^2 \\
 &= 3735.71 \text{ kg.m/s}^2 \\
 &= \underline{\underline{3735.71 \text{ N}}}
 \end{aligned}$$

The blade root area is the same with the cross-section area of the blade, which is $9.34 \times 10^{-5} \text{ m}^2$

$$\begin{aligned}
 \sigma_c &= \frac{F_c}{A_{\text{root}}} \\
 &= 3735.71 / 9.34 \times 10^{-5} \text{ m}^2 \\
 &= 39\,984\,037 \text{ N/m}^2 \\
 &= \underline{\underline{39.98 \text{ MPa}}}
 \end{aligned}$$

4.1.2.2 Calculation of the Steam Bending Stress

Firstly, what is needed is the free body diagram.

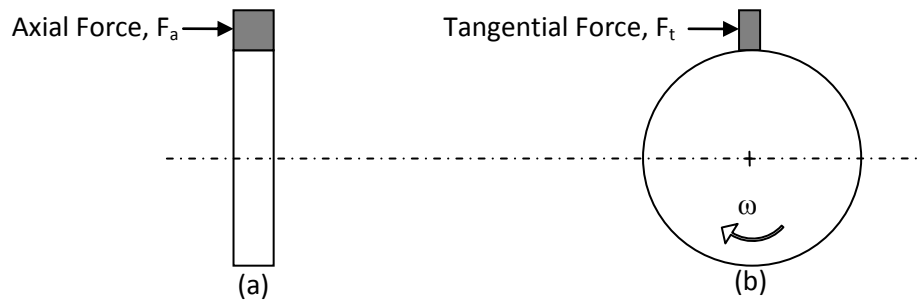


Figure 4.9: The orientation of the axial force (a) and the tangential force (b)

The free body diagram indicates that the axial force is acting on the axis of the shaft, while the tangential force is acting perpendicular to the whirl direction. This figure shows the direction of the steam which is very important in building the velocity diagram. With the aid from Figure 4.6, the velocity diagram can be drawn.

Basically, velocity diagram is like this:

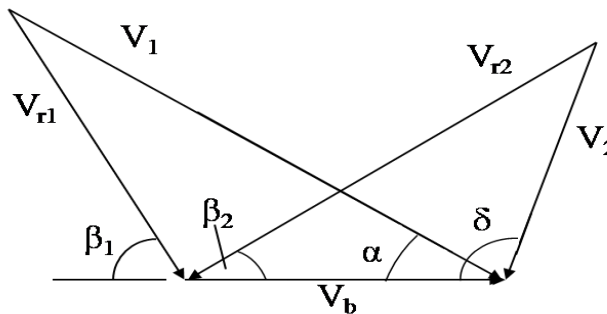


Figure 4.10: The general velocity diagram of inlet-exit steam velocity

where

- V_b = Blade velocity
- V_1 & V_2 = Inlet and exit steam velocity
- V_{r1} & V_{r2} = Inlet and exit relative steam velocity
- α & δ = Inlet and exit steam angles
- β_1 & β_2 = Angles of inlet and exit steam relative velocity

To draw a velocity diagram, there is a need to specify the basis. The basis will be the blade tangential velocity, which is obtained from equation $V_b = \omega r$, where V is the blade tangential velocity, ω is the rotational velocity and r is the length from the center of rotor to the rotor's tip.

$$\begin{aligned} V_b &= \omega r \\ &= (1289.62 \text{ rad/s})(0.2066 \text{ m}) \\ &= 266.435 \text{ m/s} \end{aligned}$$

With the aid of CATIA, the velocity diagram will be drawn. Assuming the inlet steam angle, α is 15° , β_1 is 35.748° and β_2 is 25.714° , the velocity diagram will be like this:

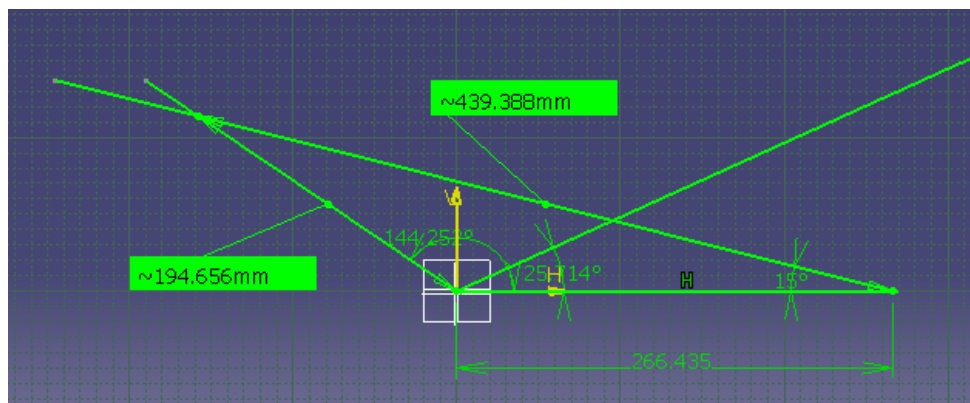


Figure 4.11: The velocity diagram of the steam, which gives the inlet velocity and relative inlet velocity

Problem arises as the exit velocity cannot be measured because of the insufficient data. However, using velocity coefficient, or blade friction factor k_b , the exit relative velocity can be calculated. The velocity coefficient is a function of the total change of direction of the steam in the blade, the blade width to radius ratio, and the relative velocity and density at blade entrance. By using equation

$$\begin{aligned} k_b &= (0.892 - 6.00 \times 10^{-5} V_{r1})^{1/2} \\ &= [0.892 - 6.00 \times 10^{-5} (194.656)]^{1/2} \\ &= 0.9383 \end{aligned}$$

Since

$$V_{r2} = k_b (V_{r1})$$

$$= 0.9383 (194.656)$$

$$= 182.656 \text{ m/s}$$

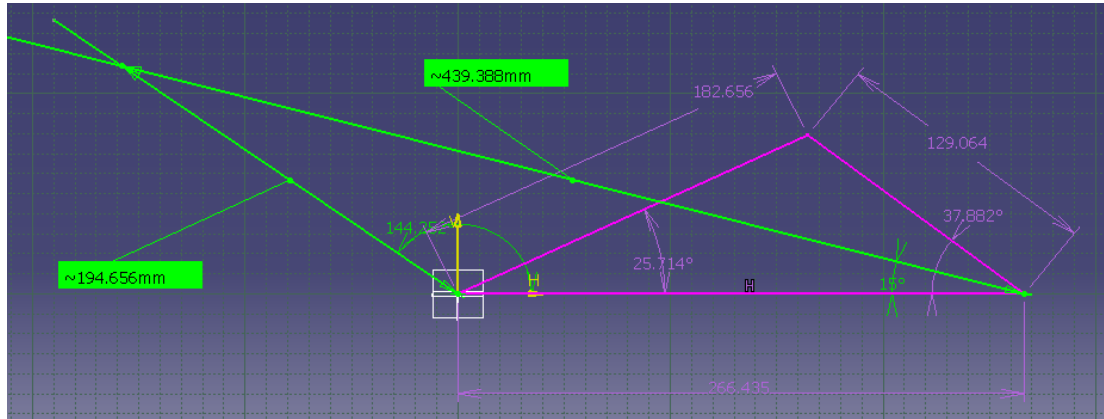


Figure 4.12: The complete velocity diagram of the steam, built with the aid of CATIA

From the velocity diagram obtained, the exit steam velocity is 129.064 m/s at angle δ is 37.882° .

Then, the velocity diagram will be resolved into axial and tangential components.

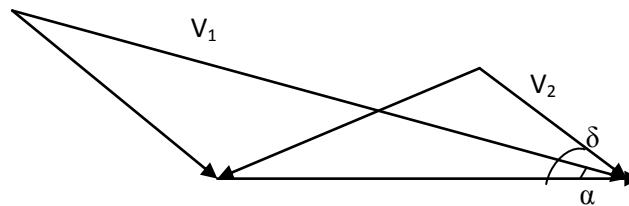


Figure 4.13: V_1 and V_2 will be resolved into axial and tangential velocities for the calculation of the steam loadings

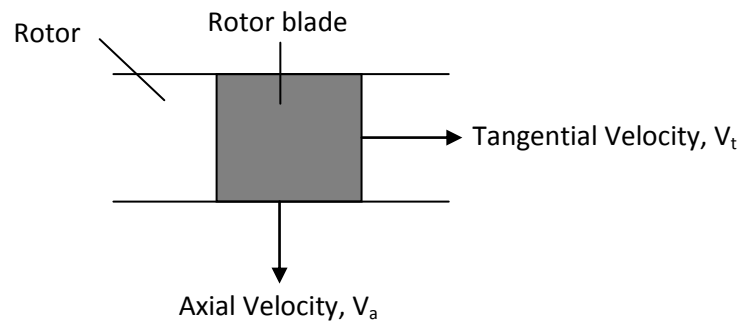


Figure 4.14: Graphical presentation of the resolved V_1 and V_2

Based on equations 2.10 and 2.11, the equations can be resolved as the following:

$$V_{1t} = V_1 \cos \alpha = 439.388 \cos (15^0) = 424.42 \text{ m/s}$$

$$V_{2t} = V_2 \cos \delta = 129.064 \cos (37.882^0) = 101.87 \text{ m/s}$$

$$\therefore \Delta V_t = V_1 \cos \alpha - V_2 \cos \delta = \underline{322.55 \text{ m/s}}$$

$$V_{1a} = V_1 \sin \alpha = 439.388 \sin (15^0) = 113.72 \text{ m/s}$$

$$V_{2a} = V_2 \sin \delta = 129.064 \sin (37.882^0) = 79.25 \text{ m/s}$$

$$\therefore \Delta V_a = V_1 \sin \alpha - V_2 \sin \delta = \underline{34.47 \text{ m/s}}$$

Now, the loading will be

$$F_t = \dot{m} (\Delta V_t)$$

$$F_a = \dot{m} (\Delta V_a)$$

Mass flow rate, \dot{m} is obtained from the turbine's operating manual. The mass flow rate for a turbine blade is

$$\begin{aligned} \dot{m} &= (27700 \text{ kg/h}) (1\text{h}/3600\text{s}) / 97 \text{ blades} \\ &= (0.07932 \text{ kg/s}) / \text{blade} \end{aligned}$$

$$\therefore F_t = \dot{m} (\Delta V_t) = (0.07932 \text{ kg/s}) (322.55 \text{ m/s}) = \underline{25.58 \text{ N}}$$

$$F_a = \dot{m} (\Delta V_a) = (0.07932 \text{ kg/s}) (34.47 \text{ m/s}) = \underline{2.73 \text{ N}}$$

The moments due to the loadings will be

$$M_t = F_t (h) = 25.58 \text{ N} (0.01571 \text{ m}) = \underline{0.4019 \text{ Nm}}$$

$$M_a = F_a (h) = 2.73 \text{ N} (0.01571 \text{ m}) = \underline{0.04289 \text{ Nm}}$$

Having the moments, the actual bending moments in the cane, which the axis is across the horns and perpendicular to a line across the horns, given θ is 10.59^0 , are

$$\begin{aligned} M_{xx} &= M_t \cos \theta + M_a \sin \theta \\ &= 0.4019 \cos(10.59^0) + 0.04289 \sin(10.59^0) \\ &= \underline{0.4029 \text{ Nm}} \end{aligned}$$

$$\begin{aligned} M_{yy} &= M_t \sin \theta + M_a \cos \theta \\ &= 0.4019 \sin(10.59^0) + 0.04289 \cos(10.59^0) \\ &= \underline{0.116 \text{ Nm}} \end{aligned}$$

Up until this point, the challenge now is to find the area moment of inertia and also the length of certain point on the turbine blade its centroid. To overcome this problem, CATIA will be used to find the I_{xx} and I_{yy} .

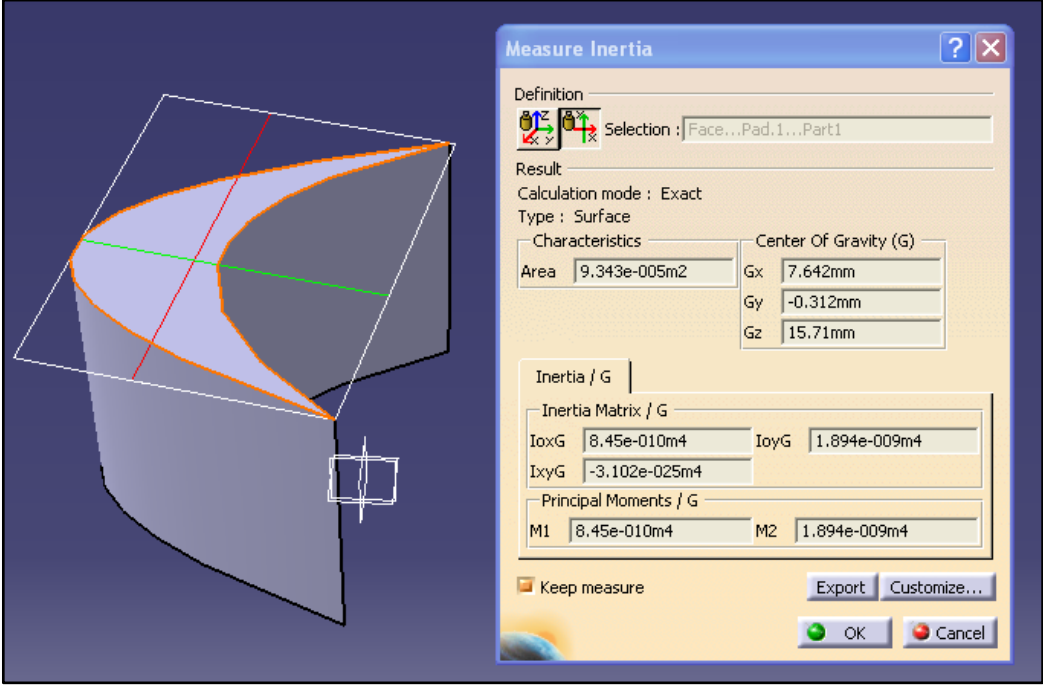


Figure 4.15: The usage of CATIA to find the area moment of inertia, I_{xx} and I_{yy}

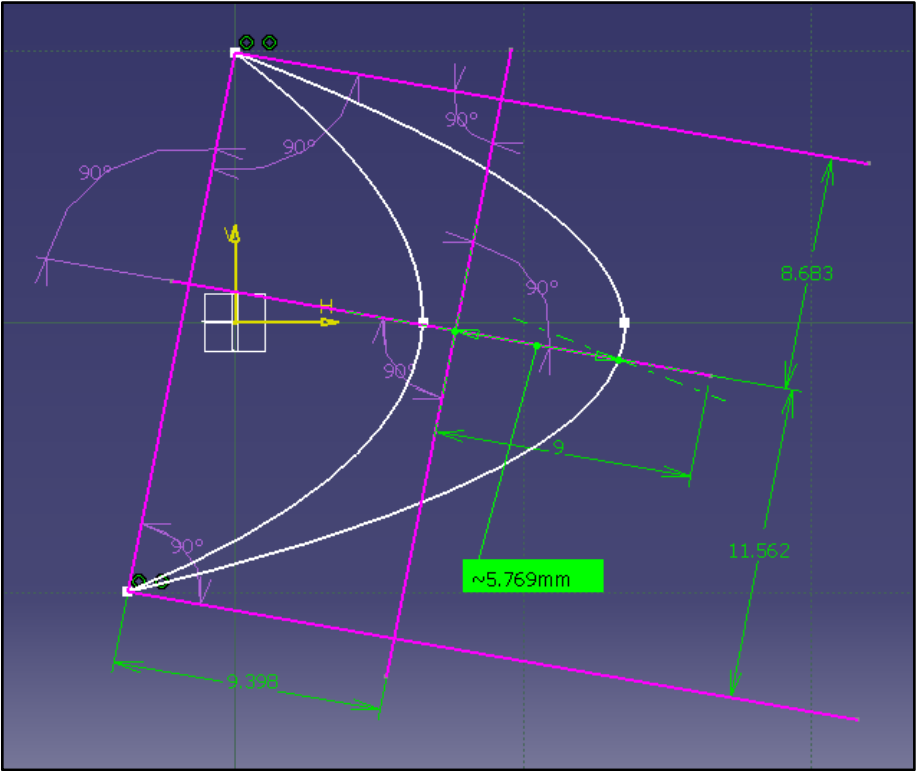


Figure 4.16: The usage of CATIA to measure the length of certain points from the centroid

$$I_{xx} = 8.45 \times 10^{-10} \text{ m}^4$$

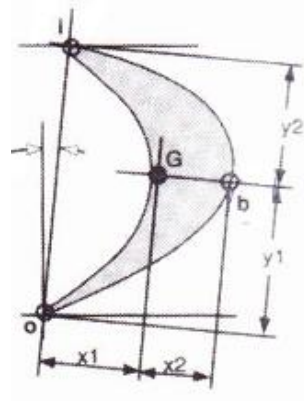
$$I_{yy} = 1.894 \times 10^{-9} \text{ m}^4$$

$$x_1 = 9.398 \text{ mm}$$

$$x_2 = 5.769 \text{ mm}$$

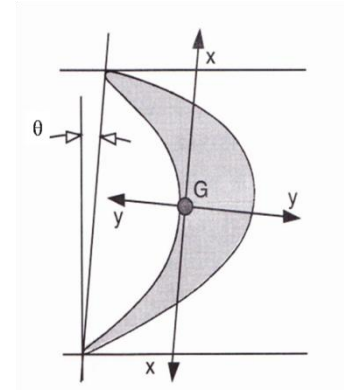
$$y_1 = 11.562 \text{ mm}$$

$$y_2 = 8.683 \text{ mm}$$



The bending stress at point i, which is at steam inlet:

$$\begin{aligned} \sigma_i &= \frac{M_{xx}}{I_{yy}} x_1 + \frac{M_{yy}}{I_{xx}} y_2 \\ &= \frac{0.4029}{1.894 \times 10^{-9}} (0.009398) + \frac{0.116}{8.45 \times 10^{-10}} (0.0086833) \\ &= \underline{3.19 \text{ MPa}} \end{aligned}$$



$$\begin{aligned} \sigma_o &= \frac{M_{xx}}{I_{yy}} x_1 + \frac{M_{yy}}{I_{xx}} y_1 \\ &= \frac{0.4029}{1.894 \times 10^{-9}} (0.009398) + \frac{0.116}{8.45 \times 10^{-10}} (0.011562) \\ &= \underline{3.59 \text{ MPa}} \end{aligned}$$

$$\begin{aligned} \sigma_b &= \frac{M_{xx}}{I_{yy}} x_2 \\ &= \frac{0.4029}{1.894 \times 10^{-9}} (0.005769) \\ &= \underline{1.23 \text{ MPa}} \end{aligned}$$

4.1.3 Meshing of the Turbine Blade in ANSYS

As discussed in methodology section 3.7.1, the meshing is done in ANSYS. The element chosen for the meshing is Hexahedral Dominant (Hex Dominant) with element size is set to 1mm. Figure 4.17 shows the meshing of the turbine blade.

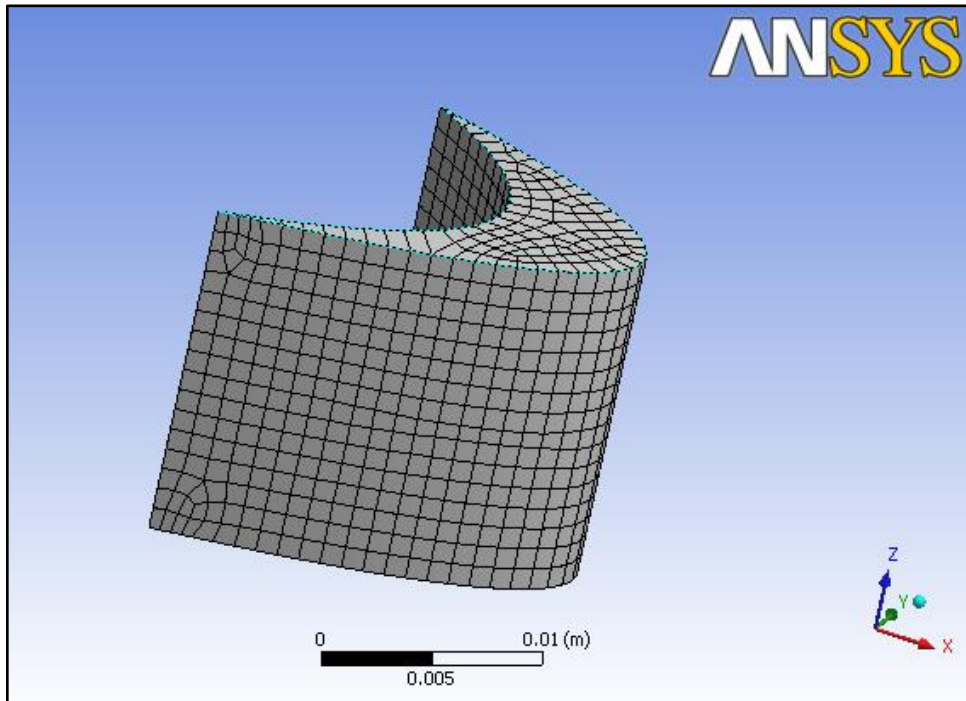


Figure 4.17: Figure shows the meshing of the turbine blade using Hexahedral Dominant method with element size of 1mm

4.1.4 Simulation of the Stress Using ANSYS

4.1.4.1 Centrifugal Stress

The first ANSYS simulation is the simulation of the centrifugal stress. The first thing to do is to specify the coordinate system (CS) of the ANSYS. Local coordinate is set as per figure 4.18, which is located at the centroid of the blade, but it is at the center of the rotor. The coordinate is (0.007642, -0.000312, -0.19089) with respect to the global CS.

The boundary condition is set to be at the lower part of the surface, which is its root attached to the rotor. The rotational velocity is then set at the center of the rotor, which is the origin y of the local CS.

Details of "Coordinate System"	
Definition	
Type	Cartesian
Ansys System	Program Controlled
Origin	
Define By	Global Coordinates
Origin X	7.642e-003 m
Origin Y	-3.12e-004 m
Origin Z	-0.19089 m
Location	Click to Change
Principal Axis	
Axis	X
Define By	Global X Axis
Orientation About Principal Axis	
Axis	Y
Define By	Default
Directional Vectors	
Transformations	
Base Configuration	Absolute
Transformed Configuration	[7.642e-003 -3.12e-0...

Figure 4.18: Set the local CS

Details of "Rotational Velocity"	
Scope	
Geometry	All Bodies
Definition	
Define By	Components
Coordinate System	Coordinate System
<input type="checkbox"/> X Component	0. rad/s (ramped)
<input type="checkbox"/> Y Component	1289.6 rad/s (ramped)
<input type="checkbox"/> Z Component	0. rad/s (ramped)
<input type="checkbox"/> X Coordinate	0. m
<input type="checkbox"/> Y Coordinate	0. m
<input type="checkbox"/> Z Coordinate	0. m
Suppressed	No

Figure 4.19: Set rotational velocity

Scope	
Scoping Method	Geometry Selection
Geometry	1 Face
Definition	
Type	Fixed Support
Suppressed	No

Figure 4.20: Set boundary condition

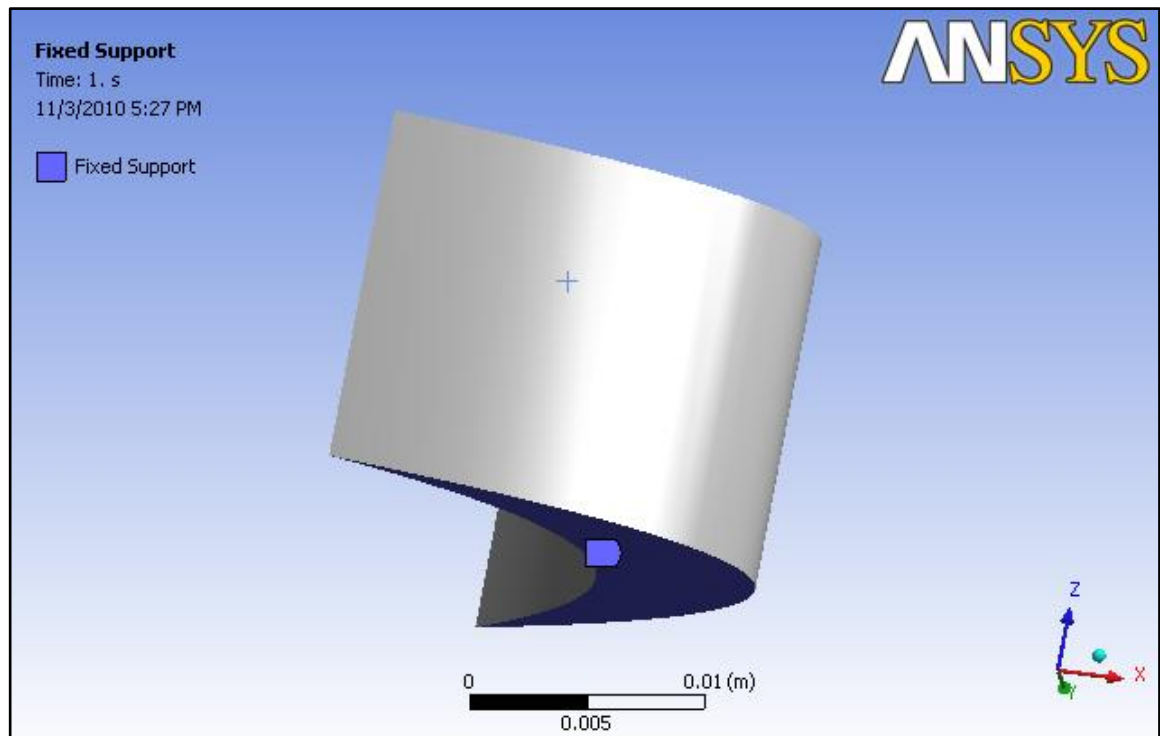


Figure 4.21: The boundary condition of the blade

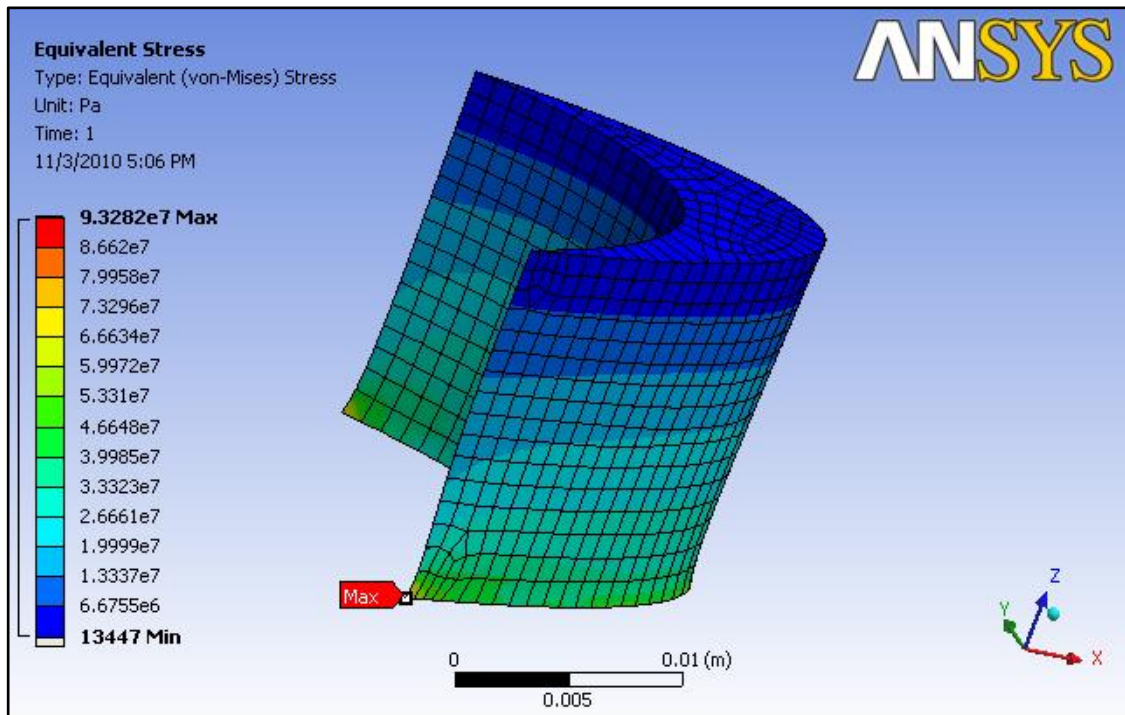


Figure 4.22: The general view of the result of simulation for centrifugal stress

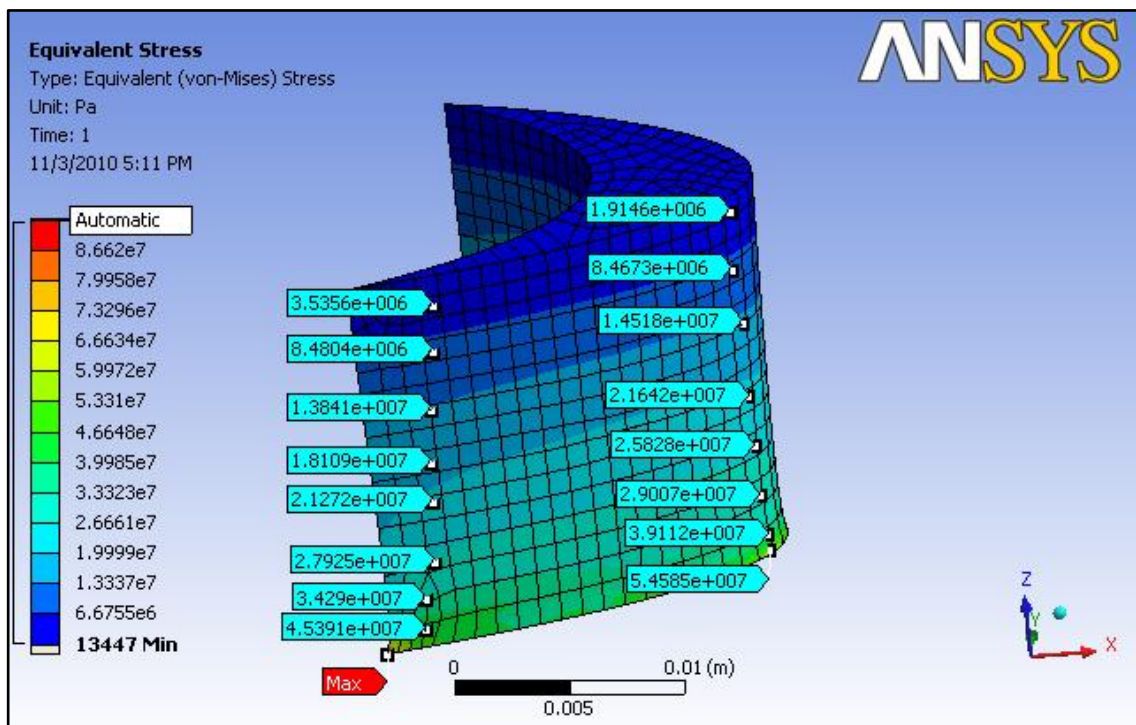


Figure 4.23: Some of readings of the blades under centrifugal stress using probe utility

Probes, from tip to root	Stress around the trailing edge	Stress around the middle
Point 1	3.5356 MPa	1.9146 MPa
Point 2	8.4804 MPa	8.4673 MPa
Point 3	13.8410 MPa	14.5180 MPa
Point 4	18.1090 MPa	21.6420 MPa
Point 5	21.2720 MPa	25.8280 MPa
Point 6	27.9250 MPa	29.0070 MPa
Point 7	34.2900 MPa	39.1120 MPa
Point 8	45.3910 MPa	54.5850 MPa

Table 4.3: The readings of the centrifugal stress around the turbine blade

4.1.4.2 Steam Bending Stress

Since the loading of tangential and axial forces are known, we can put the value of forces into ANSYS. The boundary condition applied in this problem is the same with the boundary condition in centrifugal stress simulation. The forces acted on the blade are to be applied on the other three blade surfaces. The direction of the forces is specified with respect to the local CS.

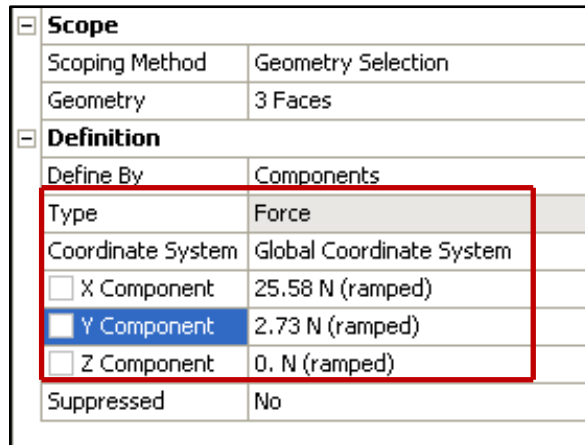


Figure 4.24: Set the loadings on the blade

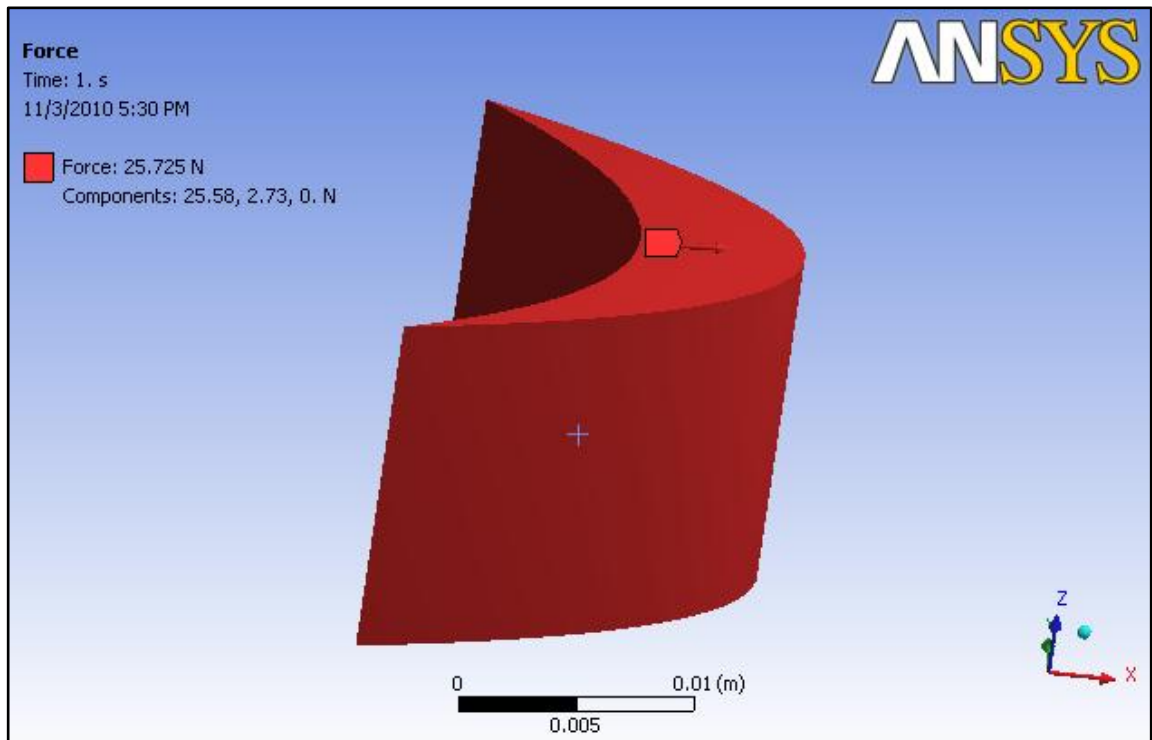


Figure 4.25: The direction of force

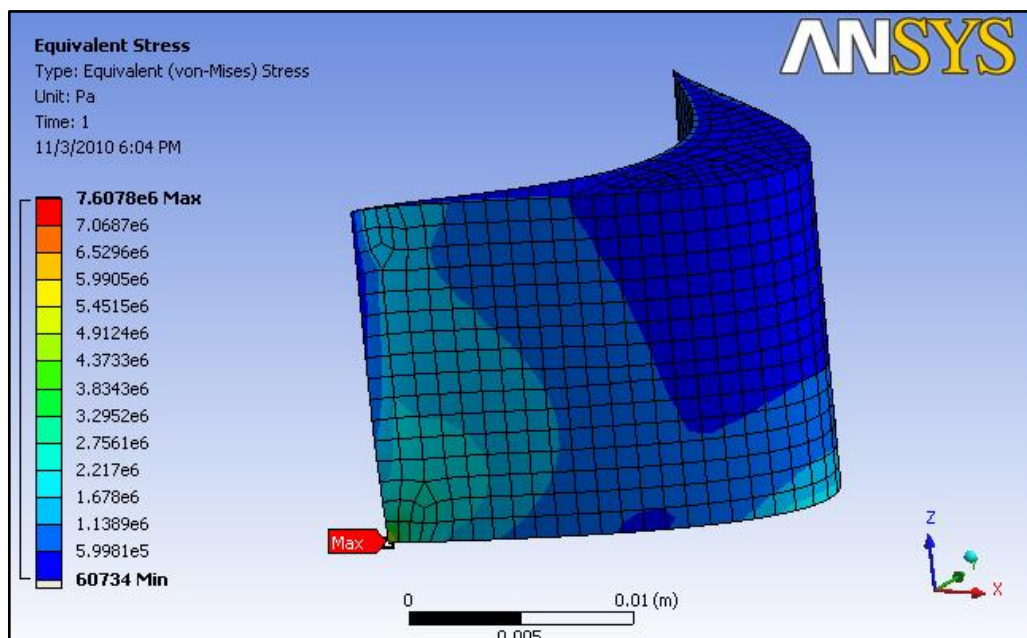


Figure 4.26: The general view of the result of simulation for steam bending stress

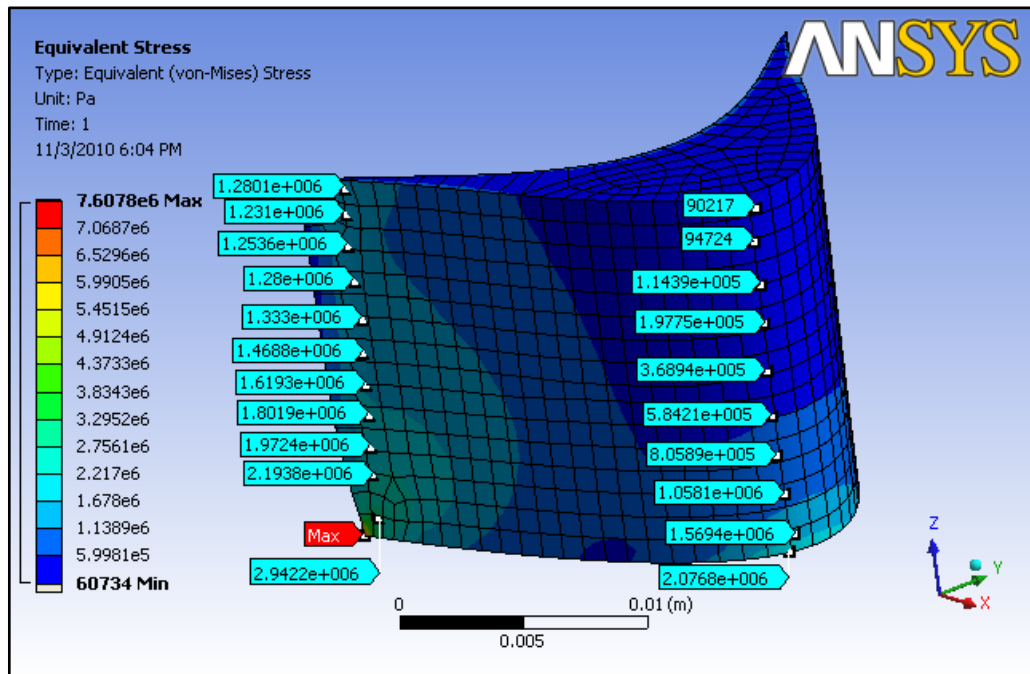


Figure 4.27: Some of readings of the blades under steam bending stress using probe utility

Probes, from tip to root	Stress around the trailing edge	Stress around the middle
Point 1	1.2801 MPa	0.0902 MPa
Point 2	1.2310 MPa	0.0947 MPa
Point 3	1.2536 MPa	0.1144 MPa
Point 4	1.2800 MPa	0.1978 MPa
Point 5	1.3330 MPa	0.3689 MPa
Point 6	1.4688 MPa	0.5842 MPa
Point 7	1.6190 MPa	0.8059 MPa
Point 8	1.8019 MPa	1.0581 MPa

Table 4.4: The readings of the steam bending stress around the turbine blade

4.1.4.3 Combination of the centrifugal stress and steam bending stress

For the last simulation, the loadings, which are the centrifugal loading and the steam force are combined to see the effect on the turbine blade. Both all the conditions specified which are the boundary condition, rotational velocity and the force acted on the specified direction is combined and simulation is run to check its result.

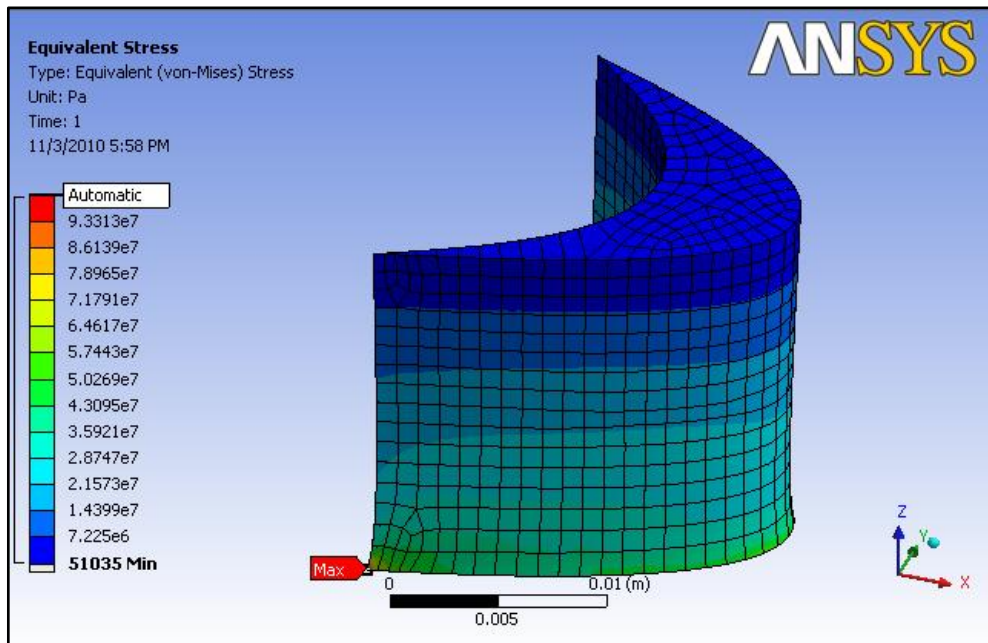


Figure 4.28: The general view of the result of simulation for the stresses combination

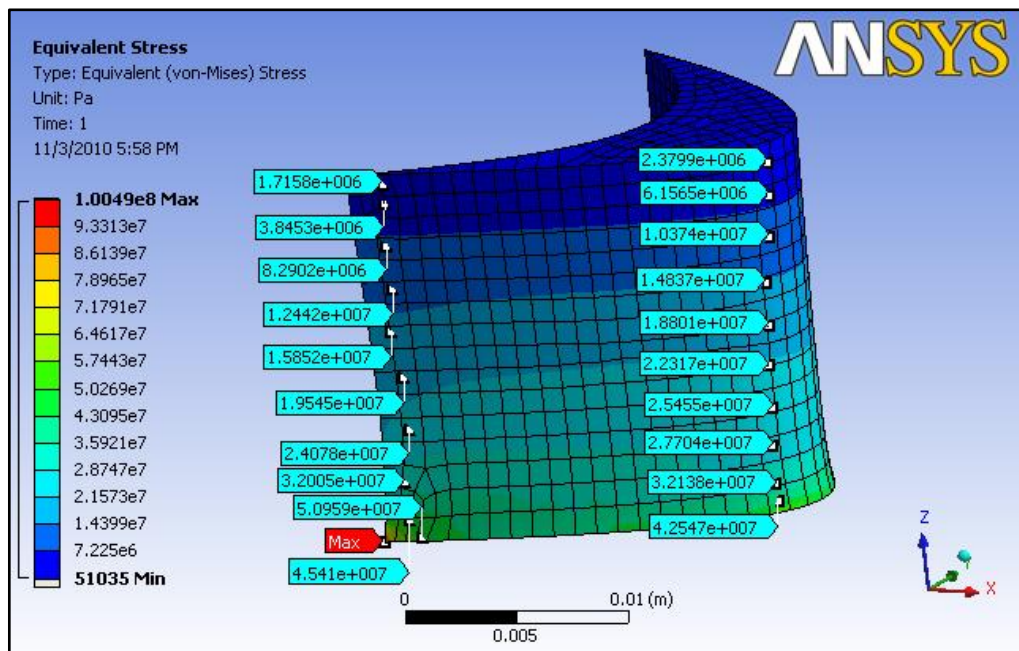


Figure 4.29: Some of readings of the blades under stresses combination using probe utility

Probes, from tip to root	Stress around the trailing edge	Stress around the middle
Point 1	1.7158 MPa	2.3799 MPa
Point 2	3.8453 MPa	6.1565 MPa
Point 3	8.2902 MPa	10.3740 MPa
Point 4	12.4420 MPa	14.8370 MPa
Point 5	15.8520 MPa	18.8010 MPa
Point 6	19.5450 MPa	22.3170 MPa
Point 7	24.0780 MPa	25.4550 MPa
Point 8	32.0050 MPa	27.7040 MPa

Table 4.5: The readings of the combination of stress around the turbine blade

4.2 Discussion

4.2.1 Three Dimensional Solid Model of the Turbine Blade

The turbine that is used in this project is KKK AFA 4 steam turbine. This turbine is the single stage impulse turbine that is operating in Kg Gajah Mill, near Pasir Salak. For the solid model, only one rotor blade is taken for the solid model in CATIA. The material properties specified are based on the material used to fabricate the blade, referring to its operating manual, which is X22CrMoV12.1. The blade shape is modeled from the measurement of the blade itself, using Vernier caliper.

4.2.2 Manual Calculation of the Stresses

Since there are two specified stress, which are the centrifugal stress and the steam bending stress, there are two different stress calculation procedures. Those procedures require the input from the turbine's operating data and also the material properties of the blade itself. The operating input required are the mass flow rate, velocity of steam get through the blade, and the gear ration and turbine output speed. The material properties are such as the blade's density, root and cross-sectional areas, and area moment of inertia.

By just looking at the free body diagram, it is known that the biggest stress concentration point is at the root of the blade. The root of the blade is located near the attachment between the rotor and the blade.

4.2.2.1 Centrifugal Stress

Centrifugal stress is the major stress that acted on a turbine blade. It is a single factor that gives the loading the most loading. In the equation, the blade length and the blade's cross-sectional area are the control factor of the centrifugal loading. Speed of rotation cannot be controlled as it depends on the amount of steam towards the turbine.

Cross-sectional area is one of the control factors of centrifugal loading. If the blade is tapered, the direct tensile stress diminishes, rapidly towards the blade tip, while the bending stress will increase with the increase of radius. Also goes for the cross-section area, the centrifugal load will not be reduced by the increase of section area since increasing the section-area simply means a greater load carried on a proportionately larger area, resulting in the same stress. Sufficient cross-section area must be provided in the blade root to withstand the maximum force at the root. The material used also must be capable to withstand the stress without failure.

4.2.2.2 Steam Bending Stress

Generally, the dominant force in steam turbine blade is the centrifugal force, however at lower rotational speed, the force that gives impact to the turbine blade is the forces from the steam flow. The impulse blade is subjected to bending stress from the tangential force by the steam flow. In this project, the turbine blade is the one without the lacing wire, so at certain points, for the ease of calculation, the blade can be regarded as a cantilever beam of variable cross-section. The bending stress will depend on the steam velocity, mass flow rate, the blade height and also the geometry of the blades.

The bending stress is maximum at blade root. Proved by calculation, the bending stress will increase by increasing the blade height. However, the bending stress diminishes with increase in the radius if the blade is tapered

4.2.3 Simulation of the Stress by ANSYS and Comparison of Results

Based on the results, the stress obtained by ANSYS mostly around the calculated values from the manual calculation. The results show that, for all cases, the stress are huge at the root area, and diminishes with radius. ANSYS is able to measure the

maximum value of stress, which is at the root of the blade. The stress measured by ANSYS is doubled as much as the stress calculated manually.

The stress concentrates at the root in a very small region. For the centrifugal stress, the manual calculation gives a reading of 39.98 MPa while ANSYS gives the maximum reading of 93.282 MPa for the centrifugal stress. This value multiplies about 2.3 times the centrifugal stress calculated manually. Also for the steam bending stress, the maximum reading given by ANSYS is 7.6078 MPa while the manually calculated steam bending stress are around 3.1-3.6 MPa, which is also half of the maximum value.

This can occur due to the existence of the sharp corner at the blade. The sharp geometry acts as the stress concentrator, which amplifies the stress. The stress is already concentrated but with the sharp edge around, the stress amplifies to a certain value at a certain point.

When compared to the manual calculation, the results from ANSYS mostly comply with the result from hand calculation. Although there are certain regions detected by ANSYS to have unmeasurable stress value, the stress is expected to be since it is located near the stress concentrator region. With computational result, it is critically important that the designer check for the validity of the results, either experimentally or by using hand calculations. Also, the finite element stress analysis results must also be evaluated by analytical solutions or if possible using hand calculation during the post-solution checks for finite element stress analysis results.

With the stress calculated and the material properties provided, the material can withstand the stress acted on it, if the operation is still within the operating condition. The maximum stress at the blade obtained from this study is 100MPa while the tensile stress for X22CrMoV12.1 is about 900MPa.

CHAPTER 5

CONCLUSION AND RECOMMENDATIONS

Stress analysis is an important part in mechanical design and stress concentration is one of the considerations that engineers need to take care of. Knowing the amount of the concentrated stress can aid in a better design and prevent unwanted accidents due to design failure.

Until now, the stress identified to be working on the steam turbine blade is centrifugal stress and fluid pressure difference. Based on the geometry and the operating data, the calculation can be proceeded.

Hand calculation is first done to obtain the stress. Catia is used to aid for the dimension calculation. Then, using all the information that have, the stress is being calculated.

Although the calculation has been done, there are a lot of rooms need to be improved. Most of the calculation is based on the rough calculation. More accurate value can be obtained in the future.

In the future, computer software will be used to aid the calculation of the stress. It will involved the 3-dimensional calculation as it will be more useful for the analysis.

The objective of this project to analyse and calculate the stresses that acted on the single-stage steam turbine blades have been achieved. Also, the simulation by ANSYS for the problems to check the reliability of the result has been carried out. It is hopefully that this project can be used as reference in the future for a better design of the steam turbine.

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APPENDICES

Appendix A Gantt Chart for Final Year Project (FYP) I

Appendix B Gantt Chart for Final Year Project (FYP) II

Appendix A

Gantt Chart for Final Year Project (FYP) I

No.	Activities / Week	1	2	3	4	5	6	7		8	9	10	11	12	13	14
1	Selection of project topic								M							
									I							
2	Preliminary Research Work								D							
	# identifying software and tools required								S							
	# research on books and journals related to project topic								E							
									M							
3	Familiarization with steam turbine blades and research on the stress that will be acted on steam turbine blades															
4	Seminar								B							
									R							
5	Familiarization with CATIA								E							
									A							
6	Continue with project work								K							
	# collect required data															
	# solid modeling using CATIA															
	# study on the manual calculation of stresses acted on the turbine blades															
7	Submission of Interim Report Final Draft															
8	Oral Presentation															

During Study Week

Appendix B

Gantt Chart for Final Year Project (FYP) II

No.	Activities / Week	1	2	3	4	5	6	7		8	9	10	11	12	13	14
1	Familiarization of ANSYS and solid modeling using CATIA								M							
									I							
2	Understanding Manual calculation of stress in the turbine blade								D							
									S							
									E							
3	Calculation of Centrifugal stress and Fluid Pressure Difference								M							
4	Seminar															
									B							
5	Simulation by ANSYS and Model Variations								R							
									E							
6	Comparison between the results of the simulation in ANSYS and manual calculation								A							
									K							
7	Oral Presentation															
8	Submission of Dissertation (Hard Bound)									After Oral Presentation						